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TECHNOLOGICAL AND ECONOMIC FEASIBILITY OF ADVANCED POWER CYCLES AND METHODS OF PRODUCING NONPOLLUTING FUELS FOR UTILITY POWER STATIONS

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The effort of the Research Laboratories on advanced-cycle power systems was directed by Mr. Ned C. Rice, Chief, Energy Conversion Systems Evaluation, and involved the following people:

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Technological and Economic Feasibility of Advanced Power Cycles and Methods of Producing Nonpolluting Fuels for Utility Power Stations

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Report J-970855-13

Technological and Economic Feasibility of Advanced Power Cycles and Methods of Producing Nonpolluting Fuels for Utility Power Stations

SUMMARY

Analytical studies have been made to identify the technical and economic factors that will govern future selection of fuel cleanup processes and advanced-cycle central power stations which, in combination, will be capable of producing electric power at the lowest possible cost while reducing substantially the emissions of sulfur oxide pollutants resulting from the combustion of high-sulfur coal and residual fuel oil. These studies were sponsored by the National Air Pollution Control Administration under Contract No. CPA 22-69-114.

The technical approach was based upon technology currently available, but possibly not yet reduced to commercial practice, as well as technology judged attainable for commercial use within the next ten and twenty years. For convenience, these are referred to as first-, second-, and third-generation systems, respectively. This approach included evaluations of current and projected fossil-fuel desulfurization and conversion processes, current and advanced-cycle central power stations, and integrated fuel cleanup and power stations. FMC Corporation performed the evaluation of fuel cleanup processes. Their evaluation included the selection of representative fuels; consideration of the form of the converted fuel; definition of the technology necessary to realize economic, large-scale conversion of fuels; and an indication of the economic benefits that could arise from potential by-product sales. The United Aircraft Research Laboratories and Burns and Roe, Inc., investigated the technical and economic feasibility of advanced-cycle central power stations designed to utilize clean fuels. Power stations investigated included those using steam, open-cycle gas turbine, combined gas and steam turbine (COGAS), closed-cycle gas turbine (utilizing helium, sulfur dioxide, or carbon dioxide working fluids), steam topping and/or bottoming (utilizing liquid metal, ammonia, or fluorocarbon working fluids), and several miscellaneous cycles. Detailed consideration was given to integration of the fuel cleanup processes and power stations. An investigation was conducted to select the combinations of fuel cleanup process and power station capable of producing lowest-cost power. Fuel cleanup process flow diagrams and power station component layout drawings are presented as are layout drawings of the integrated stations. Detailed capital cost estimates and estimates of the annual owning and operating costs are given for the selected integrated stations.

CONCLUSIONS

1. Coal and residual oil fuel cleanup processes have been identified which could reduce sulfur content to levels equivalent to, or below, those required to meet both the near-term NAPCA goal of 1000 ppm sulfur oxide pollutants in the combustion product effluent, and the ultimate goal of 50 ppm.

2. Advanced-cycle power systems have been identified which could operate at efficiencies sufficiently high to more than compensate for the higher costs of the clean fuels, thereby resulting in projected busbar power costs lower than costs from current, conventional steam stations that pollute the atmosphere.

3. The hydrodesulfurization of residual fuel oil can produce a fuel which would meet the near-term NAPCA sulfur oxide emission goal of 1000 ppm in the stack gas. However, this fuel, which can be considered a first-generation clean fuel, would be unsuitable for use in advanced-cycle power systems since the combination of metals content and residual sulfur content would cause extensive corrosion at the temperatures likely to be encountered either on heater tube walls or on turbine blades. Hydrotreating to levels of sulfur content corresponding to ultimate NAPCA emission goals would make the resulting product uneconomical for use as a power plant fuel.

4. The gasification and desulfurization of coal to meet all NAPCA sulfur oxide emission goals could be accomplished with commercially available equipment, i.e., a Lurgi-type dry-ash gasifier in conjunction with a hot-potassium carbonate scrubbing system. However, the relatively high incremental cost of clean fuel from this system (15¢ to 25¢/million Btu) would result in increases in busbar power costs from first-generation power stations.

5. Advanced second- and third-generation coal and oil cleanup systems could be developed with existing fundamental technology for high-temperature, elevated pressure gasification of coal and residual oil followed by hot-potassium carbonate scrubbing to remove all except 1% of the sulfur contained in the original fuel.

6. The clean, gaseous fuel from such systems would be suitable for use in advanced-cycle gas turbine power systems, and would have an estimated incremental cost of 10¢ to 20¢/million Btu.

7. During the next ten years, advanced-cycle power stations utilizing combined gas and steam turbine (COGAS) power systems, and burning a gasified and desulfurized fuel, have the potential of not only reducing the emissions of sulfur oxides by over 99% compared to present-day conventional power stations, but also producing electric power at costs significantly lower than those of present-day conventional power stations.

8. Technology currently being developed for advanced aircraft gas turbine engines will suffice for the development of commercial base-load utility gas turbine systems with unit capacities as high as 500 Mw. Although such equipment is not now available, its development would be evolutionary and would not require significant technical breakthroughs to reach fruition.

9. Advanced-cycle base-load gas turbine power stations burning a gasified and desulfurized fuel also have the same potential for sulfur oxide reduction and, while not as attractive as COGAS systems, would have the potential of producing power at costs which are lower than those of present-day conventional steam power stations.

10. Conventional fossil-fueled steam stations will not display significant increases in station efficiency over the next two decades. The cost of electric power from such stations, were a clean desulfurized fuel to be used, would increase by approximately 20% compared to the current cost.

11. Other advanced-cycle power systems, such as those using liquid-metal topping cycles, ammonia or fluorocarbon bottoming cycles, or closed-cycle gas turbines, while demonstrating station efficiencies comparable to or higher than those of conventional steam stations, do not offer the potential of reduced power costs.

12. Advanced-cycle power stations based upon COGAS or base-load gas turbines have the potential of significant reductions both in thermal pollution and in nitrogen oxides pollution.

13. The production of a power plant fuel based upon processes such as COED which are designed to upgrade coal or residual oil to higher-quality hydrocarbons could be potentially attractive. The economics of such processes are, however, very dependent upon the credit allowed for the by-product hydrocarbons.

RECOMMENDATIONS

1. A pilot plant consisting of a COGAS power system and a Lurgi-type coal gasification system should be constructed at the earliest possible time. Such a pilot plant would demonstrate the advantages in operating efficiency and power cost that could be realized by advanced-cycle COGAS systems. A study program with the objectives of defining the performance and identifying modifications to current equipment that would be required to construct such a pilot plant should be undertaken as soon as possible.

2. A program should be undertaken to develop an air-blown, high-temperature, elevated-pressure, entrained-flow gasifier that would produce a low-heat-content fuel gas suitable for use in advanced-cycle power stations.

3. Process improvements for the removal of sulfur compounds and particulates generated by gasification processes should be energetically pursued.

4. Research and development programs should be undertaken to hasten the appearance of advanced materials and cooling techniques which would allow the introduction of advanced-cycle gas turbines operating at turbine inlet temperatures above 2500 F in a base-load application.

INTRODUCTION

Within the next two decades, over 250 new power stations of 500 Mw or greater capacity will be required within the continental United States (Ref. 1). An estimated 35 to 50% of these will be fossil fueled (Ref. 2). Currently, the largest source of atmospheric sulfur oxide pollutants are the electric utility coal- and residual-oil-fired steam-electric stations (Ref. 3), which currently produce nearly 50% of the over 30 million tons of sulfur oxide emitted annually in this country. These fossil-fueled central power stations have the potential, if uncontrolled, of emitting approximately 30 million tons, or about 70%, of the total annual atmospheric sulfur dioxide pollutants by 1980 (Ref. 4). It is stated in Ref. 4 that the maximum rate of sulfur dioxide emission from central power stations will probably occur between 1975 and 1995 if proposed control schemes are initiated, and that a decline in the maximum rate will occur after this period due mostly to the introduction of significant nuclear generating capacity.

Currently, there are several processes in various stages of research, development, and pilot plant application for reducing sulfur oxide emissions by cleaning the central power station stack gases after combustion (Refs. 5, 6, and 7). Specifically, processes such as molten-carbonate scrubbing (Ref. 8), the Cat-Ox process (Ref. 9), dolomite injection (Ref. 10), and Wellman-Lord wet scrubbing (Ref. 11) are being evaluated. Funding by the National Air Pollution Control Administration (NAPCA) has been provided for evaluation of the relative technical and economic merits of some of these stack gas cleanup processes (Ref. 12). Concurrently, a number of fuel conversion and desulfurization processes have been developed in the laboratory to convert high-sulfur coal and residual oil into clean, low-sulfur-content fuels. However, it is apparent that the cost of processing the fuel to remove the sulfur will result in higher cost per unit of usable fuel thermal energy.

In order to offset the anticipated higher costs for desulfurized fuels, it is appropriate that there be renewed interest in methods of increasing the efficiency of central power stations. In particular, technical advances currently being developed for use in aircraft gas turbines could be applied to utility needs in numerous ways, including the use of the gas turbines alone or in combination with other cycles, or the use of projected gas turbine technology and materials in other machinery. Thus, the objective of this study was the identification of fuel desulfurization and conversion processes and advanced-cycle central power stations which, in combination, will be capable of producing the lowest-cost electric power and will reduce or eliminate emissions of sulfur oxides and other pollutants resulting from the combustion of high-sulfur coal and residual fuel oil.

This report contains three parts. Part One identifies attractive fuel conversion and desulfurization processes, Part Two identifies attractive advanced-cycle power systems, and Part Three identifies the combination of fuel processing and power systems which result in power stations having the lowest-cost electric power and reduced sulfur oxide emissions.

SCOPE OF STUDY

To achieve the objective of identifying the combinations of alternative fuel processing systems and power system types that could deliver the lowest-cost electric power in future years, a three-part study was carried out. In the first part, a review was made of current fuel cleanup and conversion processes as well as those using technology judged to be attainable within the next 10 and 20 years; the second part was concerned with defining performance characteristics and limitations of advanced-cycle power systems; and the final part dealt with the investigation of combined fuel conversion and power generation stations in commercial configuration and the subsequent selection of the optimum combinations that afford production of electricity at low cost. The technical and economic evaluations for all three parts were carried out for three time periods. First-generation systems were based upon presently available but not necessarily commercially utilized technology. Second- and third-generation systems were based on projected technology judged by UARL to be attainable within the next 10 and 20 years, respectively.

The performance and economic evaluations of both the fuel processing systems and the power systems were based upon the ability to supply fuel for and generate 1000 Mw of power within prescribed sulfur oxide emission goals. The near-term or first-generation goal set by NAPCA was to limit emissions from the combustion of 3.5% sulfur coal or 2.5% sulfur residual oil to the equivalent of 1000 ppm SO_2 in the effluent. The intermediate, or second-generation, goal was the equivalent of 200 ppm, and the ultimate, or third-generation, goal was the equivalent of 50 ppm. The coal-fired station was to be in the Great Lakes region (southern)

Illinois was selected), while the oil-fired station was to be on the east coast (a Hudson River location was chosen).

Fuel processing systems were initially sized to supply fuel to a nominal 1000-Mw power station at 40% station efficiency, and capital and annual owning and operating costs were determined. Several fuel conversion processes were eliminated because of their inability to meet the projected sulfur emission goals, and others were eliminated because of unfavorable economics. Concurrently, the performance of various advanced-cycle power systems was being evaluated. Several power systems were eliminated from further consideration based upon performance alone. For the majority, however, preliminary cost estimates were made, and the projected costs of generating power were estimated. Based upon a comparison of the projected busbar cost of power, further elimination of power systems was made.

Having identified the most favorable power systems, their performance characteristics were then used as a basis of modifying fuel cleanup processes to form an integrated fuel cleanup and power station. More refined estimates of the cost of electrical power were then made, and those systems offering the lowest-cost power were identified.

PART ONE

FUEL DESULFURIZATION PROCESS STUDIES

SUMMARY

An investigation was undertaken to estimate the technical performance and economic feasibility of processes for the conversion of high-sulfur coal and residual oil to desulfurized fuels. A review of available literature was made to establish present desulfurization technology which formed the basis for projections of future technology. Flow sheets are presented for those processes judged to have the potential of meeting sulfur oxide emission goals projected for the next two decades. Economic characteristics of these selected processes are presented, and the estimated costs of the resulting desulfurized fuels are compared.

The processes considered for coal conversion were: physical processing, including mechanical separation; chemical processing, including deashing, coal extraction, and pyrolysis; gasification, including the Lurgi-type dry-ash process, partial-oxidation with slagging, molten salt process, and the carbon dioxide acceptor (dolomite) process; and chemical desulfurization processes, including a liquid acceptor process. The processes considered for oil conversion were: hydrotreating, gasification, coking with coke gasification and desulfurization, and catalytic oxidation of sulfur.

The above coal and oil conversion processes were evaluated for three technological levels: technology currently available, although not necessarily reduced to commercial application, technology judged to be attainable within ten years, and technology judged to be attainable within twenty years. These levels of technology are called first-generation, second-generation, and third-generation technology, respectively.

For the purposes of this study, a power system with a nominal capacity of 1000 Mw was selected as the basic unit for comparing costs of different fuel processes. Thus, the size of a fuel processing plant would be roughly defined as being sufficient to supply fuel to a 1000-Mw power system. However, the actual size of the fuel processing plant would depend upon the power system efficiency which, in turn, would be a function of the type of fuel being delivered, the particular power cycle selected, and the time frame for which it is proposed. For these reasons, the preliminary analysis of fuel processing schemes was based on a nominal 1000-Mw power system operating at approximately 40% efficiency. Subsequently, the fuel processing system costs were adjusted to reflect the specific capacities required by the selected power systems.

For a liquid fuel, the nominal capacity for a fuel processing system to supply a 1000-Mw power system would be about 34,000 bbl/day. The economics of a central fuel processing plant serving a number of power plants was also investigated. This central processing plant would supply the fuel equivalent of 5000 Mw and have a nominal size of 170,000 bbl/day.

The size of plant delivering a gaseous fuel would depend on the station efficiency, as previously noted, but would consume on the order of 7000 tons/day of coal and 30,000 bbl/day of oil, both of which would subsequently be gasified.

The geographic sites selected for the power station are discussed in Appendix A. For fuel processes which gasify coal or residual oil, the fuel processing plant is assumed to be adjacent to the power station and integrated with it as required (see Appendix B). For processes which desulfurize Venezuelan residual oil, a fuel processing plant adjacent to the power plant would not be required, but was so assumed. In addition, a large hydrodesulfurization plant to serve multiple East Coast power plants was investigated. This plant would either have an East Coast location or be located in Venezuela, where low-cost natural gas provides an inexpensive source of hydrogen as well as process fuel.

Initially, goals were set for sulfur emissions in ppm of SO_2 in the stack effluent; 1000 ppm for 1970 technology and 50 ppm as the ultimate goal. However, this measure depends on the amount of excess air required for combustion and would create an artificial advantage for gas turbine systems. Another criterion for sulfur emissions is the weight of sulfur per unit of fuel consumed, grams per million Btu, adjusted to a constant amount of excess air. This criterion was used for all comparisons of alternative fuel desulfurization processes. Sulfur emissions of selected processes are also compared per unit of power generated, grams per kwhr, in order to include improvements in pollution abatement due to efficiency improvements in energy conversion.

Initially, these criteria were set at 450 g/million Btu for first-generation processes (1000 ppm SO_2); 90 g/million Btu for second-generation processes (200 ppm SO_2); and about 20 g/million Btu for third-generation processes (50 ppm SO_2). Included in the above estimates was the equivalent amount of sulfur emitted from the fuel processing, so that the total sulfur emission was taken into account. Preliminary work was based on these guidelines.

As the study developed, however, it became apparent that the above guidelines were unrealistic. Certain processes were inherently capable of removing a certain amount of sulfur, and it made no sense to try to operate them to remove much less sulfur, or much more sulfur, simply to satisfy the arbitrary guidelines. Therefore, the guidelines were dropped, and each process was analyzed on the basis of removing all of the sulfur that could reasonably be removed, regardless of time period for which the process was considered. The ultimate goal of about 20 g/million Btu was kept in mind, however, as a qualitative guideline, when processes were judged.

SYNOPSIS OF FUEL DESULFURIZATION PROCESS STUDY RESULTS

The gasification of coal or residual oil in air-blown, elevated-pressure, and temperature gasifiers followed by liquid scrubbing in hot-potassium carbonate absorption towers appears to offer the most promising method of delivering a fuel low in sulfur to advanced-cycle power systems at reasonable cost. For first-generation use, only the Lurgi-type fixed-bed, nonslagging gasifier would be applicable. However, later-generation fuel processing systems could use slagging, entrained-flow gasification vessels which could further reduce both the investment and annual costs of the system. The costs of the fuel delivered to the power system would be a function of power plant efficiency, sulfur level in the fuel, time period considered, and several other factors; however, the estimated incremental costs for clean gaseous fuel range from 10¢ to 20¢/million Btu over the cost of the untreated fuel used as feedstock.

The sulfur oxide emissions from the overall fuel processing system-power system complex would be the sum of the emissions from each. It has been projected that the equivalent of a total minimum sulfur content of 6.5 g/million Btu (equivalent to 15 ppm SO_2 at 10% excess air) could be realized by a hot-potassium carbonate scrubbing system used in conjunction with a modified Claus sulfur recovery plant in which the tail-gas from the Claus plant is recycled to the scrubbing system. The technology to do this is currently available.

Residual oil can be hydrodesulfurized to meet first-generation sulfur emission goals but would not be a suitable fuel for most advanced-cycle power systems because of the unacceptably high metals content in the fuel. The metals would hasten corrosion of heater tube walls or turbine blades at high temperatures. Further hydrotreating or catalytic treatment could reduce sulfur and metals content, but the properties of the residual oil would be so changed that it would be uneconomical for use as a fuel.

Fuel treatment processes such as COED which produce upgraded hydrocarbon by-products and a char could be attractive. The char would be gasified in a slagging gasifier, desulfurized in the liquid scrubbing system, and sent to the power system. The liquid hydrocarbons would be sold, and the gaseous hydrocarbons could be blended with the gasified char or used internally in the fuel process. System capital costs would be higher than those for straight fuel gas production processes; however, the annual operating costs could be attractive if reasonable credits were allowed for the liquid hydrocarbon by-products.

The use of high-temperature sulfur removal techniques which preserve the sensible heat contained in the gasified fuel does not appear to offer significant advantage over the low-temperature liquid scrubbing systems. The projected fuel costs for a dolomite process with in situ sulfur removal at 1500 F are essentially the same as the costs for the straight gasification and liquid scrubbing system, but the dolomite process does not have as high a sulfur removal potential as does the liquid scrubbing system.

DETAILED FUEL DESULFURIZATION PROCESS STUDIES

Section I - Review of Fuel Desulfurization Technology

A review of available literature on fuel desulfurization was made in order to establish the technology now available and to project the time of appearance of advanced technology. Having established the relative levels of technology, initial elimination of unattractive processes was done based upon the potential of a given process to meet the sulfur oxide emissions goals for the particular time period in which it was projected to appear.

It became readily apparent that the gasification of coal in conjunction with a desulfurization process was the only method that could yield a power plant fuel of acceptably low sulfur content at reasonable costs. Residual oil could be hydrodesulfurized to meet first-generation emission goals, but hydrodesulfurization processes to meet more stringent goals would so upgrade the residual that it would not be economic for use in power plants. Thus, gasification of residual oil appears to be the most attractive method of obtaining a clean fuel.

This section contains reviews and discussions of desulfurization technology and defines those fuel processing systems that have the potential of supplying low-sulfur fuel to advanced-cycle power systems. The characteristics of the specific residual oil and coal used during the fuel conversion analyses are described in Appendix C. Cost estimates for fuel conversion processes were prepared according to the method of cost analysis described in Appendix D.

COAL DESULFURIZATION TECHNOLOGY

Many options are available for the production of low-sulfur clean fuel from coal. These options can be broken down into two main categories: physical methods and chemical methods. Physical methods include sulfur and ash removal by washing, by flotation, and by solvation. Chemical methods include pyrolysis and gasification. All these options and methods were surveyed to determine optimum methods for reaching first-generation objectives.

Physical Methods

Washing and flotation, two physical methods of sulfur removal, remove only inorganic sulfur associated with the ash constituents of coal. These methods cannot remove organic sulfur and therefore would not be applicable to many of the coals of

Illinois, Indiana, and western Kentucky where organic sulfur predominates. Experimental work by the Bureau of Mines (Refs. 13 and 14) on coals of Pennsylvania, Ohio, West Virginia, and eastern Kentucky shows that most coals containing more than 3% total sulfur would not meet first-generation sulfur oxide emission goals after simple washing. For this reason, simple washing and flotation methods of inorganic sulfur removal were not considered further.

A new physical method, solvation, is currently being developed (Refs. 15, 16, and 17). This method evidently will work on almost any coal except anthracite. The coal is slurried with a suitable solvent and pressurized to about 1000 psi pressure. The slurry under pressure is then gradually heated to about 800 F. Under these conditions, nearly all the organic constituents of the coal are dissolved, leaving behind the ash and inorganic sulfides, which are then filtered out of the solution under pressure. Hydrogen is recycled in this process to facilitate the dissolution by partial hydrogenation of the coal solution. Some of the hydrogen combines with the organic sulfides to produce H_2S gas which is subsequently removed from the recycle gas.

One variation of this general process is known as the P-M Process and is now under development by the Spencer Chemical Co. A review of the economics of these processes has been published in Ref. 18. A principal advantage of this process is that the final product, a partially desulfurized and de-ashed fuel, has the form of the original coal (e.g., in lumps), which permits cheap transportation via railroad hopper cars. Another advantage is its low melting point, in the order of 250 F, which permits conversion on site to a liquid fuel which can be pumped at about 430 F.

Liquid fuels, similar to heavy residual oil, which meet first-generation sulfur criteria have also been produced by solvation of coal. However, this work has not progressed to a level which would allow its introduction as a source of utility fuel within the time span considered for first-generation systems.

To meet more stringent second- and third-generation sulfur guidelines, the coal extract would have to be hydrotreated, but the hydrotreated extract was judged to be more valuable as a petroleum or petrochemical feedstock than as a fuel. Thus, straight solvation does not appear to be a feasible candidate process for producing a clean fuel.

Chemical Methods

Pyrolysis

Pyrolysis processes produce tar, medium-Btu gas, and char or coke by the destructive distillation of coal in the absence of air. Such processes could produce a medium-Btu gas and a hot, highly reactive solid product eminently suitable for gasification. A combined pyrolysis and gasification process could be economically attractive if by-product credits taken for the tar produced covered the cost of pyrolysis and parts of the cost of gasification.

Assuming pyrolysis processing costs of $2.50/ton of coal (exclusive of raw coal cost) and a tar value of $2.00 to $2.50/bbl (32 to 40¢/MM Btu), a tar yield of over 20 wt % would be required to justify pyrolysis (Ref. 19). Typical commercial pyrolysis processes such as coking and low-temperature carbonization yield less than 10% tar (Ref. 20). The COED multistage, fluidized-bed process has demonstrated better than 20% tar yields experimentally (Ref. 19). However, this process, which would not be commercially available until the second-generation time period, does not directly produce a low-sulfur fuel suitable for advanced-cycle power system use. Rather, the resulting char from the process would require gasification and desulfurization.

Gasification

Commercial Gasification Processes

The gasification of coal is an existing commercial technique to change the form of coal from solid to gas. Other commercial techniques are available to desulfurize the resulting fuel gas to meet the clean fuel criteria.

The basic gasification process consists of reacting steam with coal at elevated temperatures. The main reaction occurring is:

$$C + H_2O \rightarrow CO + H_2 \qquad (1)$$

This reaction is endothermic, the heat for this reaction being supplied by combusting a portion of the coal in air or oxygen:

$$C + \frac{1}{2} O_2 \rightarrow CO \qquad (2)$$

$$C + O_2 \rightarrow CO_2 \qquad (3)$$

In addition, the water-gas shift reaction

$$CO + H_2O \rightarrow CO_2 + H \qquad (4)$$

occurs.

Many options are available for the gasification of coal. Heat for the process can be supplied autothermally wherein reactions (1), (2), and (3) take place simultaneously, or externally, by combusting a portion of the coal to supply heat through a wall or a heat transfer medium. The heating value of the product gas can range from 100 to 500 Btu/ft^3*, depending on whether air or oxygen and internal or external heating is employed. Gasification can be conducted at atmospheric or elevated pressures in countercurrent, cocurrent, or fluidized-bed gasifiers.

Since the fuel processing plant and power plant would be located together, there would be no need for a high-Btu/ft^3 gas, i.e., long-distance transportation of the gas would not be required. Thus, there appears to be no justification for the use of oxygen in an autothermal gasifier. Oxygen requirements to supply heat for reaction (1) range from 0.65 to 1.0 lb O_2/lb coal (Ref. 20). Assuming a gasification efficiency of 85% and an oxygen cost of $6/ton, the cost of oxygen would range from 15 to 20¢/million Btu of fuel gas. The use of oxygen would result in capital cost savings in the sulfur removal portion of the process due to a decrease in the volume of gas handled. However, assuming annual capital charges of 14% plus 5% for maintenance, a capital cost saving of $35 to $45/kw would be required to justify the use of oxygen. This cost saving is of the same order as the capital cost of the gasification processes which use air.

The processed fuel gas must be at elevated pressure for use in some power systems. However, elevated pressures would be required for the fuel processing system in order to reduce the size of the sulfur removal equipment. Since the fuel gas volume is approximately twice the volume of air required for gasification, compression costs would be lowered significantly if gasification takes place at elevated pressure. Also, specific gasification rates, i.e., gas produced per cubic foot of reactor volume, are increased by increased pressure. Thus, gasification at elevated pressures appears to be desirable.

Countercurrent, cocurrent, and fluidized-bed gasifiers have been used commercially (Ref. 21). Countercurrent gasifiers with downflow of coal and upflow of gases have the advantage of high thermal efficiency and flexibility in gasification rates. Their major disadvantages are low gasification rates, relatively small capacities, and the formation of tar which makes gas purification and waste heat recovery difficult. Cocurrent gasifiers (up- or downflow of both coal and gases) have relatively low thermal efficiencies unless expensive waste heat recovery equipment is employed. Gasification rates are higher than those of countercurrent gasifiers because higher temperatures can be employed. A major advantage is that no tar is formed, making gas cleanup and waste heat recovery easier. Fluidized-bed gasifiers have intermediate properties. They can be scaled up to relatively large sizes. However, turndown ratios for these units are small due to the necessity of maintaining a minimum fluidizing velocity through the unit.

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* The term ft^3, when referring to gases, is understood to mean standard cubic foot, i.e., at 60 F, 14.7 psia, unless otherwise designated.

A summary and comparison of some of the more promising commercial gasification processes surveyed are given in Table 1. Of these processes only the Lurgi dry-ash process is carried out at elevated pressures. For this reason the Lurgi process was chosen for detailed analysis as a first-generation fuel conversion process. The Wellman-Galusha process, an atmospheric-pressure, fixed-bed process, was selected for a less-detailed analysis to determine the effect of pressure on the economics of first-generation, fixed-bed processes. The Bamag-Winkler fluidized-bed process, and the Koppers Totzek, B&W-DuPont, and Rummel Single Shaft processes, all cocurrent processes, are run at atmospheric pressure. In the subsequent section on first-generation desulfurization processes it is shown that high-pressure processes are more economical than atmospheric-pressure processes, especially when the requirements for gas cleaning and gas turbine power cycles are considered.

Advanced Gasification Technology

Advanced technology for coal desulfurization is considered to be that technology which would be feasible for use during second- and third-generation time periods. Two classifications of advanced processes were considered: advanced coal gasification processes and advanced gasification processes which would be coupled with processes that produce valuable by-products from coal. Advanced gasification processes would be carried out at high temperatures and pressures, and thus would allow high specific gasification rates to be achieved. Such processes would utilize cocurrent flow within the gasification vessel and would require the recovery of the sensible heat in the hot gasifier off-gas to achieve satisfactory efficiencies. Sulfur would be removed from the off-gas either by low-temperature liquid scrubbing or by high-temperature adsorption on solids such as iron oxide or calcined dolomite as is described in subsequent paragraphs. Processes which could produce valuable liquid hydrocarbon by-products and residual solids to be gasified include pyrolysis and solvation.

Advanced gasification processes could provide a usable fuel for an advanced-cycle gas turbine power system. The fuel which would result from such a process would be a low-Btu (100 to 200 Btu/ft^3) gas supplied at pressures suitable for use directly in the combustion chamber of the engine. A low-Btu, producer-gas-type fuel would be formed by the autothermal gasification of coal with air and steam. High-pressure producer gas is most economically supplied by gasifying at elevated pressure levels, levels which correspond closely to the turbine inlet pressure. Thus, any advanced gasification process selected would operate at elevated pressures.

Present-day, high-pressure gasification technology is limited to a fixed-bed, countercurrent process, the Lurgi dry-ash process. Temperatures in the Lurgi process must be kept below the ash fusion temperature of the coal. For this reason, gasification rates are relatively low, and capital costs of the reaction vessels are relatively high, being equivalent to nearly half the total capital cost for the equipment required to make clean fuel from coal. To achieve higher gasification rates, temperatures in excess of 2200 F must be employed. Under these conditions, the coal ash would melt and become slag. Thus, a fixed- or fluidized-bed process could no longer be used, and an entrained or cocurrent flow process would need to be employed.

Various high-temperature, cocurrent flow gasification processes have been surveyed. Gasifier configurations vary, but the basic processes, rates, efficiencies, and costs appear to be the same. Reactant flow can be tangential as in the Kopper's Totzek process; upflow as in the B&W-DuPont, Rummel Single Shaft, Ruhrgas Vortex, and BCR processes; or downflow as in the Texaco, USBM, and IGT processes (Table 1). In all cases, from two- to three-second residence times are required for 90 to 100% carbon conversion, and gasification temperatures range from 2200 F to 2500 F. Oxygen or air requirements must be sufficient to supply heat for preheating feeds, endothermic reactions, and making up heat losses. Steam requirements must be such that the reactant temperature is kept within the 2200 F to 2500 F range by the endothermic steam-carbon reaction. Off-gases from these reactors generally reach equilibrium with respect to the water-gas shift reaction. Capital costs for such gasifiers are on the order of $4 to $5/kw, or roughly 15% of the capital required for gasification plants to make clean fuel gas from coal. Unlike the first-generation, fixed-bed process where gasification is the controlling capital cost, air compression and heat recovery costs would be the controlling costs for advanced gasification processes.

Unlike the countercurrent Lurgi process where the fuel gas exits the gasifier at 950 F and contains less than 8% of the heating value of the coal as sensible heat, the fuel gas would exit from advanced, cocurrent gasifiers at 2200 F to 2500 F and contain from 15 to 20% of the heating value of the coal as sensible heat. Recovery of this heat is desirable from an efficiency standpoint, but it must be balanced against the capital charges for heat recovery equipment. Increases in process efficiency can be directly translated into savings in both annual fuel costs and annual capital costs. Figure 1 shows the allowable capital cost of heat recovery equipment as a function of the percent of the higher heating value of coal recovered as sensible heat. From Fig. 1 allowable costs for heat recovery equipment for second-generation plants would range from $9.50 to $12.00/kw, and for third-generation plants from $7.50 to $11.00/kw. These values are about 25% of the capital cost for the advanced plants.

Two options are available for sensible heat recovery from hot fuel gas. In the first, the sensible heat would be used to preheat the air, steam, and coal feeds to the gasifier. In the second option, the heat would be used to raise high-quality steam for power generation. If heat were recovered in preheating feeds, less air would be required for gasification, resulting in lower air compression costs. Also, analysis showed that lower overall costs would result if all the power were to be generated by the high-efficiency, advanced-cycle power system, rather than in a combination of the advanced-cycle system and a steam system using the sensible heat in the fuel gas. The two options are compared for second- and third-generation systems in Table 2. As shown in this table, the total capital and operating cost savings that could be realized if the sensible heat were used to preheat the feed streams would be on the order of 0.25 mills/kwhr.

The preceding arguments apply only to those processes in which the fuel gas must be cooled prior to sulfur removal. Thus, it applies to all liquid scrubbing processes such as the potassium carbonate process. Considerably capital savings in heat recovery equipment could be realized if high-temperature sulfur removal schemes were employed. Two high-temperature sulfur removal processes, the iron oxide process and the calcined dolomite process (Refs. 22 and 23), were investigated for this purpose. In the iron oxide process, iron oxide is reacted with hydrogen sulfide at 600 F to form iron sulfide. Iron oxide is regenerated by roasting, yielding by-product sulfuric acid. In the calcined dolomite process, calcined dolomite is reacted with hydrogen sulfide at 1500 F to form calcium sulfide. Regeneration of the calcium sulfide is effected by reaction with carbon dioxide and steam, yielding by-product sulfur.

The Kellogg Molten Salt Gasification Process was also investigated. The sulfur is removed in this process by the reaction of sodium carbonate with hydrogen sulfide to give sodium sulfide which is then regenerated with carbon dioxide and water to yield by-product sulfur. Experimental evidence (Ref. 24) indicates that almost none of the sulfur originally in the coal is found in the fuel gas.

Gasification with By-Product Recovery

Additional cost reductions might be achieved by using processes which would produce valuable liquid hydrocarbons as by-products along with residual char suitable for gasification. Such processes must produce liquid hydrocarbons in yields high enough to cover the cost of additional coal processing. Two such advanced processes which were investigated are the COED multistage, fluidized-bed pyrolysis process, and the Project GASOLINE (Ref. 25) solvation process. In the COED process, coal would be pyrolyzed in four fluidized beds in series, each at a successively higher temperature. Reported yields for this process are: oil - 20%, gas - 21%, char - 54%, and liquid - 5%. After debiting a portion of the gas which must be converted to hydrogen to upgrade the oil product, the gas yield would be about 18%. The char would then be gasified in a high-temperature, cocurrent gasifier and desulfurized to yield a clean fuel gas. Economics for the COED process depend heavily on the credit taken for by-product oil. At an oil credit of $3 to $4/bbl, fuel costs would be comparable to those for a straight gasification process.

The Project GASOLINE process would have a higher oil yield than the COED process. Coal would be dissolved in a coal-derived solvent which would then be hydrotreated to upgrade it to a synthetic crude oil. Reported yields for the process are: oil - 34%, gas - 12%, char - 41%, and heavy extract - 13%. All of the gas and heavy extract produced in the process must be converted to hydrogen for use in oil hydrotreating. The char from the solvation process would be gasified and desulfurized to provide a clean fuel gas.

OIL DESULFURIZATION TECHNOLOGY

Processes Based on Hydrodesulfurization

Hydrodesulfurization of heavy gas oil, such as vacuum gas oil and coker gas oil, has become a common practice as a pretreatment of catalytic cracking feedstocks and as a method of producing low-sulfur fuel oils. The processes that can be employed for hydrodesulfurization of heavy gas oil will have been extended from either the middle distillates hydrodesulfurization processes or the residual oil hydrocracking processes.

Commercial Hydrodesulfurization Technology

Some commercial processes offered for hydrodesulfurization of heavy gas oil are:

Process	Licensor
Gulfining	Gulf Research and Development
Hydrofining	Esso Research and Engineering
Isomax	Chevron Research; Universal Oil Products (UOP)
Trickle Hydrodesulfurization	Shell Development
HGO - Unicracking	Union Oil Company
Unifining	Union Oil Company; UOP

All of these processes are similar. A fixed-bed reactor is employed, and the most commonly used catalyst is cobalt molybdenum on an alumina base. Other catalysts may be used such as oxides and sulfides of iron, nickel, cobalt, chromium, molybdenum, copper, manganese, and tungsten; metals such as sodium, potassium, lithium, calcium, zinc, aluminum manganese, iron, nickel, cobalt, and copper; or compounds such as molybdates, thio-molybdates, or thio-tungstates, either alone or in combination with other catalysts. Feedstocks generally have a boiling range of 600 F to 1100 F with reaction conditions typically being:

Average bed temperature	600 to 850 F
Pressure	400 to 2000 psig
Liquid hourly space velocity (LHSV)	0.5 to 5 hr^{-1}
Hydrogen-to-oil ratio	500 to 10,000 ft^3/bbl.

During desulfurization, some denitrogenation and aromatic saturation occur. Some cracking is unavoidable, and production of fuel gases, gasoline, and light gas oil generally occurs. The gravity of the full range product will increase about 5.0 deg API. Hydrogen consumption is typically in the range of 350 to 450 ft^3/bbl.

Atmospheric residual oil may be desulfurized by hydrotreating by four methods as shown in Fig. 2. In process A, the residual oil undergoes a vacuum distillation and the vacuum gas oil is hydrodesulfurized and blended back with the vacuum bottom. The vacuum gas oil can be readily desulfurized by any of the processes described above to about 90%, but since only a portion of the total feedstock is treated, an overall sulfur reduction of about 40% is typically achieved.

For further reductions in sulfur content, additional treatment is required. This is shown in process B, where the bottoms of the vacuum distillation are further processed by de-asphalting. The de-asphalted oil is hydrodesulfurized and blended back with the asphalt fraction and vacuum gas oil. Additional sulfur may be removed as shown in process C. Here, the asphalt fraction (a vacuum bottom) is coked, with subsequent hydrodesulfurization of the coker gas oil and blending.

In order to achieve high sulfur removals by hydrotreating, complex operations consisting of a number of distinct, small units must be employed, a condition which is generally very costly. To circumvent these difficulties, it is possible to use process D, wherein the entire residual oil charge is hydrodesulfurized.

While this process eliminates some of the problems associated with the indirect hydrodesulfurization methods, it does introduce some considerable new technical problems. Due to the large quantity of asphaltenes in residual oils and high metal contents such as nickel and vanadium, rapid coking and deactivation of the catalyst occur. This is an especially serious problem with a Venezuelan residual oil.

There are basically two kinds of processes used for the direct hydrodesulfurization of residual fuel oil. They are the fixed-bed processes such as RCD-Isomax, Gulf-HDS, Esso, and Institut Francais de Petroles hydrodesulfurization, and an ebullated-bed reactor process such as H-Oil (Hydrocarbon Research and Cities Service).A more-expensive catalyst, which needs frequent replacement, is used in the fixed-bed process. In the ebullated-bed process, a less-expensive catalyst is used, and regeneration is not required since a portion is withdrawn and added during operation to maintain a constant catalyst activity level. However, the operation of the ebullated-bed process is somewhat more complicated, and a higher hydrogen consumption results in order to achieve the same desulfurization as for the fixed bed.

When attempting to employ a hydrodesulfurization process charging a high sulfur residual fraction, it is usually necessary to operate at conditions that remove only about half the sulfur present in the stock in order to provide a commercially feasible catalyst life. The reason for this is that the sulfur present in the feedstock occurs in a great variety of compounds, some of which are very resistant to hydrotreating. The more readily removed forms of sulfur are the thiols, cyclic thiols, aromatic thiols, and sulfides. These compounds are generally lower boiling and are found in the gas oil after a distillation. It is for this reason that desulfurization of vacuum gas oil can approach 100%. The more difficult to remove sulfur compounds are thiophenes, benzothiophenes, dibenzothiophenes, thienothiophenes, and higher thiophenes. These are high boiling and frequently combined in complex ring structures which also contain metals difficult to remove, even at extreme conditions of pressure and temperature where extensive cracking and other wide reactions can occur.

Thus the use of high severity conditions for hydrodesulfurization of a residual stock has been considered unfeasible because of the short catalyst life when removing a high percentage of sulfur. Also, the employment of milder conditions will not bring about the removal of sufficient sulfur to provide the desired low sulfur product.

Another detriment is the metals content in the residual oil. These metals may occur as oxides or sulfides present as particles in the oil, or as oil soluble salts. Generally, however, the metals appear as organometallic compounds of high molecular weight, such as metallic porphyrin and its derivatives. The metals tend to deposit on the hydrodesulfurization catalyst, increasing in concentration as the process continues. Thus, catalyst deactivation in residual desulfurization processes can occur from deposition of metals, as well as from coking.

There are several approaches to prolong catalyst life, although none as yet appear to be commercially successful. They are:

1. Improvement of catalyst in fixed-bed reactor
2. Employment of ebullated-bed reactor
3. Employment of a slurry process
4. Use of a homogeneous catalyst
5. Prior removal of metallic contaminants.

Most present-day commercially available processes such as RCD Isomax, Gulf HDS, Esso Hydrodesulfurization, and IFP Hydrodesulfurization employ the fixed-bed reactor, and these processes are essentially the same as far as the process scheme is concerned. The only major difference among these processes is the catalyst composition. UOP has tried new catalyst materials such as the decomposition products of phosphomolybdic acid and nickelous sulfate carried on equimolar quantities of alumina and silica, nickel acetylacetone and molybdenum acetylacetone impregnated in alumina-silica, as well as others. Gulf Research and Development Co. has used fine control over catalyst pore radius as a means of extended fixed-bed catalyst lifetimes.

In other work by Gulf Research and Development Co. the catalyst life that is used in the fixed-bed reactor is claimed to be substantially improved by contacting the asphaltic-free stock with the catalyst wherein the metalliferous hydrogenating component of the catalyst is in a substantially unsulfided state, and by sulfiding the catalyst by the asphaltic-free sulfur-containing hydrocarbon oils before the residual oils are cut in.

Another approach in maintaining the high catalyst activity is the employment of a so-called ebullated-bed reactor in which the oil and hydrogen are passed upward through the bed of catalyst at a velocity sufficient to maintain the catalyst in a state of random motion. This process was named H-Oil and commercialized by Hydrocarbon Research, Inc. and Cities Service Research and Development Co. The following advantages of the ebullated bed are claimed (Refs. 26, 27, and 28):

1. The catalyst can be added and withdrawn while the unit is operating. In the ebullated state, the catalyst behaves very much as in a gas-solid fluidized bed and can be made to flow into and out of a reactor. Activity can then be maintained at a steady, desired level by controlling the rate of catalyst addition.

2. The reactor is maintained in the ebullated state by recycling liquid from the top of the reactor to the bottom. The isothermal system created eliminates the need for elaborate quench systems for temperature control. The temperature gradient from top to bottom is reported seldom to exceed 5 F.

3. Any fine solids in the feed will pass harmlessly through the reactor system causing no operating problems, since the catalyst is in a state of constant motion.

4. The low pressure drop in the reactor permits the use of smaller catalyst sizes than the ones used in the fixed-bed reactor. These smaller catalysts are considered more effective than the usual 1/8 in. or larger particles.

In comparison with the fixed-bed reactor, the ebullated bed may have such disadvantages as a high degree of erosion, catalyst loss due to the large amount of oil circulation to keep the catalyst bed ebullated, and relatively large quantities of catalyst necessary to make proper contact between oil and catalyst active site (US patent 3,262,874).

In the H-Oil process, any conventional hydrodesulfurization catalyst such as cobalt molybdate carried on alumina can be employed. The first H-Oil plant, which has operated since March 1963 at Lake Charles, La., for Cities Service and has a capacity of 2500 bpsd, employs extruded pellets of 1/32 in. diameter and $\frac{1}{4}$ in. length. Details of the ebullated-bed reactor are reported in US patents such as 3,197,286, 3,197,288, and 3,151,060 which are assigned to Hydrocarbon Research and in 3,124,518 and 3,188,286 which are assigned to Cities Service.

The third approach to maintaining the high catalyst activity is the employment of the suspension process in which a finely divided hydrodesulfurization catalyst is slurried in the feed oil, and the mixture of oil and catalyst is charged to the reactor at which hydrogen is contacted. This kind of process has been patented by Esso Research & Engineering Co. (US patents 2,717,855 and 2,907,476). For slurrying, the catalyst is in a powder form of usually 100- to 400-mesh size. The disadvantage of the slurry process is that it may tend to result in a high degree of erosion, which will cause plant maintenance to be expensive. Furthermore, this type of operation has the disadvantage of having a relatively small amount of catalyst intimately admixed with relatively large quantities of asphaltenic material, since it is difficult to suspend more than a small percentage of catalyst within the feedstock.

The fourth approach which is tried in the patents assigned to Esso Research and Engineering Co. (US patents 2,636,841; 2,657,175) is the use of a homogeneous catalyst. A homogeneous catalyst process means a system wherein the catalyst is completely miscible in the material being treated. Thus, a solid reaction product, formed as a result of the catalytic treatment, exists in a separate phase and cannot inactivate the catalyst by forming a layer of surface coating, as is the case with heterogeneous catalysts. These homogeneous catalysts claimed in the patents are the halogen-promoted iron group metal carbonyls such as nickel carbonyl and cobalt carbonyl, boron halide complex, and stannic chloride or titanium chloride. This process seems to have a disadvantage in that it needs a rather complex catalyst recovery and regeneration system.

The fifth approach for the maintenance of high catalyst activity is the removal of ash and metal contaminants before the residual oil is charged to the hydrodesulfurization reactor. In a patent assigned to Esso Research & Engineering Co. (US patent 2,717,855), the residual oil is de-ashed by first preheating to 700 to 900 F and then heat-soaking the stock for 0.5 to 5 hours to agglomerate the ash-forming constituents. A patent to American Cyanamid (US patent 2,917,456) claims the usage of porous coke in the oil slurry to be a suitable adsorbent for the tarry matter formed during refining in the ebullated-bed reactor. The residual fuel oil is first charged to a bed of bauxite to remove vanadium and sodium from the charge stock before it is fed to the hydrodesulfurization reactor in a patent assigned to British Petroleum (British patent 735,782). Instead of bauxite, Raschig rings or other shaped filler bodies of ceramic substance, metals, bleaching earth, alumina, or synthetic silicates are also claimed for removal of ash in a patent assigned to Badische Anilin & Soda Fabrik (German patent 939,278; US patent 2,893,697).

In the commercial operation of residual oil hydrodesulfurization plant, the degree of sulfur removal seems to be limited to 80% and is preferably 75%. Above a desulfurization of approximately 80%, hydrogen consumption increases sharply because of increased hydrocracking (Refs. 29 and 30). The hydrogen consumption to produce up to 75% desulfurization is in the range of one to five molecular weight of hydrogen per atomic weight of sulfur removed.

Regardless of which commercial process is used, the hydrodesulfurization conditions to obtain up to 75% desulfurization are believed to exist in the following ranges (Refs. 31, 32, and 33):

Average Bed Temperature	700 to 850 F
Pressure	500 to 3000 psig
Hydrogen-to-oil ratio	5000 to 10,000 ft^3/bbl
LHSV	0.5 to 3.0 hr^{-1}

It is reported that impurities such as nitrogen and light hydrocarbons normally present in hydrogen-rich streams have no adverse effect either on the catalyst or on the process. If these inerts are permitted to build up to high concentrations, the effective hydrogen partial pressure will be reduced, and it will be necessary to raise the system pressure to offset this effect.

The overall life of the catalyst that is employed in RCD Isomax is reported as 35 to 45 bbl/lb of catalyst for atmospheric residual oils and 20 to 30 bbl/lb of catalyst for vacuum residues (Ref. 34). Generally, the catalyst life is proportional to the amount of metal contents in the feed, and in high-metal-content oils the catalyst life is reported as 25 bbl/lb for the atmospheric residue (metal contents of 190 ppm). The catalyst is generally loaded in the reactor for the operation for one year (Ref. 34) with regeneration every 3 to 4 months in the fixed-bed reactors. In the ebullated-bed operation of the 2500-bpsd H-Oil plant, the catalyst is added at the top of the reactor by way of a hopper, a valve, retainer pod, and a pair of block valves and removed from the bottom through a similar arrangement at the rate of 2% each day while the unit is on stream.

The hydrodesulfurization of residual fuel oil results in a fuel with a pour point reported to be 15 F to 50 F lower than that of the feedstock. The viscosity is also reduced by 3 to 7 centistokes. Nitrogen that is contained in the feed converts into ammonia by hydrogenation with conversion rates ranging from 30% to 50 wt %.

Advanced Hydrodesulfurization Technology

Advanced hydrodesulfurization technology is considered to be a technology which appears to be technically feasible for second- and third-generation fuel conversion processes. Some of the processes previously mentioned fall into this category; e.g., sulfur removal by chemical methods, improved catalysis, homogeneous catalysis, and bacteriological sulfur removal. While some of these processes have been studied in the laboratory and pilot plant tests and no doubt could be considered technically feasible for future time periods, there is also another criterion that they must satisfy, that of a high percentage of sulfur removal.

Without gasifying the residual oil, the most likely technological advancement appears to be in improved catalysis for hydrodesulfurization, wherein the total residual oil feedstock may be desulfurized to low sulfur levels. Improvements in catalysis may lead to longer catalyst life, less extreme operating conditions, lower capital investment, and decreased hydrogen consumption. While research is going on in the area of improved hydrodesulfurization catalysis, rough cost projections indicate that this is not a promising avenue.

Venezuelan residual oil is a difficult oil to treat by hydrodesulfurization because of the high metals content and the nature of the sulfur compounds which are present. A comparison of hydrodesulfurization costs is given in Ref. 35. This comparison gives the costs for desulfurizing Venezuelan residual oil to various sulfur levels and the cost for desulfurizing a Middle Eastern residual oil which is easier to hydrodesulfurize. These costs are summarized in Table 3. While the basis of the study leading to the costs given in Table 3 is slightly different than that used in the present study, the costs are close enough to make a valid comparison. The cost for total residuum desulfurization to 92% sulfur reduction has been estimated at 43.6¢/million Btu which is in agreement with the numbers shown in Table 3. Thus, those numbers can be used to estimate fuel costs as a function of percent sulfur in the fuel down to sulfur levels of 0.3%.

The figures for a Middle Eastern residuum are for comparative purposes in order to show what can be done with an "easy" residuum. Optimistically, improvements in catalysis can be projected, as can improved processing which might permit desulfurization to low levels (about 0.3%) at an incremental cost of 3.5¢/million Btu/% sulfur, as in the case of Middle Eastern oil. This would bring the cost of desulfurized Venezuelan residuum (0.3%) down from 47¢/million Btu to about 39¢/million Btu, equivalent to an incremental cost of only 9¢/million Btu.

In order to determine what future improvements might be made, several persons knowledgeable in the area of catalysis for hydrodesulfurization were contacted and asked to make projections of improvements in hydrodesulfurization technology. Those contacted, and their affiliation, are:

Dr. L. A. Nicolai
Esso Research and Engineering

Dr. W. C. Pfefferle
Engelhard Industries

Mr. J. F. Wright
UOP, Process Division

Mr. J. Rogers
American Cyanamid

Mr. D. Jackson
Nalco Chemical Co.
Catalyst Division

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Some replies were more optimistic than others in projecting improvements in catalysis and hydrodesulfurization technology. However, there was a general consensus that even if it were eventually feasible to hydrodesulfurize Venezuelan residual oil to very low sulfur levels, the product oil would be too valuable to use as a fuel for power generation because of its low sulfur content and its altered properties. It would thus command a premium price as a starting feed for more valuable liquid petroleum products.

Gasification Processes

The Kellogg Molten Salt Coal Gasification Process may conceivably be operated on a residual oil (Refs. 36, 37, and 38). This process, however, is being developed primarily to produce pipeline gas from coal by using molten sodium carbonate as a heat transfer agent and catalyst for the endothermic carbon-steam and exothermic carbon-oxygen reactions needed for coal gasification. Presumably, sulfur is removed during the gasification process by the sodium carbonate melt which contains the ash. The melt would have to be regenerated. Other processes, also conceived primarily for making pipeline gas from coal, are the CO_2 Acceptor Gasification Process (Ref. 39) and similar processes described in Refs. 40, 41, and 42. In these processes, coal, or presumably, residual oil, is gasified in the presence of dolomite, which reacts with sulfur at about 1600 F. The dolomite is then regenerated at about 2000 F.

Another possibility would be to gasify residual oil via a partial oxidation process with subsequent removal of the hydrogen sulfide formed. The hydrogen sulfide removal may be effected by conventional scrubbing or by advanced processes which will remove H_2S at high temperatures, thereby conserving the sensible heat of the fuel gas and eliminating expensive heat recovery equipment. High-temperature hydrogen sulfide removal may be accomplished by the use of hot ferric oxide at about 600 F or by dolomite at about 1600 F.

The partial oxidation of liquid hydrocarbons is a well-developed technology, and there are numerous plants in operation working on a wide variety of feedstocks. The partial oxidation process was first developed by Texaco Development (Refs. 43, 44, and 45). General descriptions of the Texaco process at various stages of development have also appeared in the literature, e.g., Refs. 46 and 47. Two other companies, Shell and Montecatini, have also made contributions to the technology of noncatalytic partial oxidation of hydrocarbons and have processes for license.

Generally, the partial oxidation process has been developed for the production of synthesis gas or hydrogen, but it is very flexible in its operating conditions. Feedstock and steam are preheated and mixed before entering the reaction chamber, which is refractory lined. Oxygen is usually used as the oxidizing agent, but air or enriched air may also be used. When air is used it is usually preheated, although the preheat of oxygen is optional. The partial combustion proceeds at 2000 to 2800 F with the exothermic combustion reactions being balanced by the endothermic reforming reactions. The operating pressure is usually set by the end use of the gas but may be as high as 100 atm. The hydrocarbon charge is converted to carbon monoxide, hydrogen, carbon dioxide, and some methane. The relative amounts of carbon monoxide and hydrogen depend on the feedstock composition, the steam-to-feed ratio, and the oxygen-to-feed ratio. Typical figures for a heavy fuel oil are a steam-to-feed ratio of 12 ft^3 of O_2/lb feed. Using oxygen, the product gas contains about 45% each of carbon monoxide and hydrogen and 5% carbon dioxide, the remainder being hydrogen sulfide, carbonyl sulfide, and nitrogen that was introduced with the oxygen. Performance using air is similar except that the product gas is diluted with nitrogen.

Heavy oils, high in metals content, were found to attack refractory linings of the reactor. This problem was overcome by adjusting operating conditions so that free carbon was formed. It is believed that the free carbon sequesters the metallic ash constituents which would otherwise deteriorate the refractory linings. The free carbon is subsequently scrubbed out of the product gas and recycled to the reactor so that there is no net carbon production. Ash accumulations are periodically withdrawn from the bottom of the reactor.

Another possibility for producing a clean fuel from residual oil is to coke the residual oil entirely. The coker gas oil is assumed to be a by-product for which no credit is taken. The energy for generating power comes from the coke, which has to be desulfurized. Technology for coke desulfurization is similar, in concept, to technology for coal desulfurization.

Desulfurization of Residual Fuel Oil by Other Processes

Various methods have been developed for treating or extracting the simple sulfur compounds in petroleum light distillates by means of chemicals, clays, and solvents. But few of them can be applied to the heavy fractions such as residual fuel oil in which the main sulfur compound is a thiophenic type. The chemical and clay treating methods developed for use on mercaptans either fail to desulfurize thiophenic type compounds or require large amounts of expensive chemicals. Acid extraction and solvent extraction remove the entire sulfur-bearing molecule in addition to some hydrocarbons and would result in large losses of material if applied to high-sulfur residual oils. Disposal of the extracted material also would be a difficult problem.

Many processes have been proposed to desulfurize the crude oil or residual oil by means other than hydrogenation and gasification. These processes can be classified as follows: (1) bacteriological treatment, (2) metallic reactions, (3) chemical treatment, and (4) oxidation. None of these methods could be considered first-generation methods.

Desulfurization by Bacteriological Treatment

It has been known for 25 years that sulfur compounds in petroleum are attached by microbes. The sulfur-oxidizing bacteria are a heterogeneous group of microbes with inorganic sulfur compounds playing a major part in their metabolism. In other respects, they generally bear little relationship to one another, although in nature they often occur together in ecosystems based on the sulfur cycle. A particular example is a polluted pond, with sulfate reduction taking place in the depths and reoxidation of sulfide and sulfur occurring near the surface. For practical purposes, the sulfur-oxidizing bacteria fall conveniently into four subgroups: (1) the sulfur-oxidizing bacteria such as Thiolocillus, (2) the sulfate-reducing bacteria such as Desulfovibio, (3) the colorless sulfur bacteria such as Beggiatoa, and (4) the colored sulfur bacteria such as Chromatium ahd Chlobium (Ref. 48).

Extensive laboratory work has been done by the Texaco Development Corporation on bacteriological desulfurization. However, no attempt at commercial application of this process has been made.

A patent assigned to Esso Research and Engineering Company in 1961 (US patent 2,975,103) has been introduced worldwide because it has shown some possibility of commercial desulfurization by the bacteriological approach for the first time. According to this patent, the sulfur-containing hydrocarbons make contact with an aqueous culture of aerobic bacteria to convert the organic sulfur compounds to inorganic sulfur compounds in the presence of oxygen. The desulfurization can be carried out either in a packed column or in a stirred vessel. When a stirred vessel is used, the bacteria culture is impregnated on an alumina carrier and added to the vessel as a water slurry. Air is bubbled through the mixture during desulfurization. The operation can be carried out between 85 and 95 F. The proportion of water to oil present during the desulfurization ranges from 0.1 to 0.2 volume of water per volume of oil. The optimum pH depends on the bacteria used and generally is in the range of 6.3 to 8.5. After the desulfurization, the mixture of aqueous culture of bacteria and oil is charged to the settler where oil phase and water phase are separated. Lime or barium oxide is added to this settler, to precipitate the sulfate formed during the desulfurization as calcium sulfate or barium sulfate. Since the number of bacteria increased excessively during the desulfurization, excess bacteria are removed from the settler.

In the above process, it is reported that more than 90% of the sulfur contained in the petroleum may be removed under aerobic conditions in less than 10 minutes contact time. Among the bacteria that may be employed are Thiophyso Volutans, Thiobacillus, Thioxidans, and Thiobacillus Thioparus.

Since the above patent was issued in 1961, no further information has been published on the bacteriological treatments. Therefore, it is impossible to evaluate the future possibility of commercialization at present from a single source, particularly when the operating conditions and the result of desulfurization are not shown for the heavy fraction of petroleum.

Metallic Reactions

It has been known for a long time that metals and their compounds, particularly oxides, react with sulfur compounds to form stable sulfides and that these reactions can be used in petroleum refining. But metallic reactions have been considered uneconomical compared with hydrodesulfurization mainly because of the difficulty of regenerating the metallic compounds. Recently, several patents have been published, claiming that these processes are now economical in the desulfurization of crude oil and residual fuel oil in comparison with hydrodesulfurization.

In the patents assigned to Shell International Research (British 958,470; German 1,276,266), the alkali metal ferrites such as sodium or potassium ferrite are employed to desulfurize heavy hydrocarbon oils at temperatures of 730 to 810 F, pressures between 14 and 105 atm, and contact times of 2 to 15 minutes. The sulfur compound in the oil is converted to the alkali metal sulfide, which in turn can be regenerated into the original ferrite by calcination at 1400 to 1850 F. According to the patent example, the degree of desulfurization is limited to about 60%.

Pyrophoric-reduced iron powders that are obtained by reducing an iron ore at 500 to 800 C in a hydrogen stream are claimed in a patent to Yawata Iron & Steel (British 1,076,755; US 3,284,345). The contact between the reduced iron powders and oil is kept for 30 minutes to one hour, but the degree of desulfurization is about 30 to 50%.

Another method of desulfurization, put forth by Tenneco Chemicals, is to grind a mixture of the sulfur-bearing fuel and the powdered iron in a mill. The grinding promotes contact of sulfur with iron, allowing the formation of iron sulfide, which can be separated out by centrifugation (Refs. 49 and 50).

The use of potassium hydroxide at 700 F for removing sulfur from residual oil has been reported by Esso (Ref. 51) (British patent 1,183,695), indicating sulfur removals of about 60%.

Chemical Treatment

Several patents have been issued recently on the desulfurization of the heavy fraction by the chemical agents. The chemicals that are claimed in the patents include maleic acid, succinic acid (Japanese patent 41-8071), molten potassium hydroxide containing 7 to 25% water (British patent 913,729), molten potassium hydroxide containing 10 to 15% water in the presence of oxygen (US patent 3,383,304), molten cesium hydroxide containing 10 to 20% water (British patent 913,730), and a mixture of lime or limestone and clay (US patent 2,970,956). In all these processes, however, the degree of sulfur removal seems to be limited to about 60%.

Another chemical treatment that desulfurizes crude petroleum at 1 atm and 90 F has been reported (Ref. 52). An unspecified treating agent is used to remove sulfur, including thiophenic sulfur. The chemical agent is subsequently regenerated.

Oxidation

In a patent assigned to British Petroleum (British patent 949,392), the heavy fraction of petroleum is desulfurized first by oxidizing it with various oxidizing agents and subsequently by subjecting it to a thermal decomposition by rapid heating to remove sulfur, mainly in the form of sulfur dioxide. Various kinds of oxidizing agents may be used, such as peroxides, hydroperoxides, peracids, chlorine, nitrogen oxides, ozone, and molecular oxygen or air. Oxidation is carried out at 260 F to 350 F for 0.5 to 20 hours, followed by thermal decomposition at 550 F to 750 F. The maximum degree of desulfurization of the atmospheric residue that appears in the patent is about 30%, and this fact, together with slowness of reaction, may show that the oxidation process is far from commercialization at the present stage.

In another patent (British patent 972,803), the above oxidation process is combined with the hydrodesulfurization process to improve the degree of desulfurization.

SULFUR RECOVERY AND REMOVAL FROM FUEL GAS

If coal or residual oil is gasified, the sulfur would appear in the off-gas principally as H_2S with small but important quantities of COS (carbonyl sulfide). There is a wealth of technological data available for the removal of H_2S from hydrocarbon gases, largely due to the development of the natural gas industry during the past 30 years. Since most natural gases must be transported through pipelines many miles from their source to the point of consumption, it is essential that all acid gases, principally CO_2 and H_2S, be removed, since they are a potential source of corrosion of the pipelines.

Liquid Scrubbing System

Amine Scrubbing Systems

There are several principal methods for removing CO_2 and H_2S from natural gas. These methods are based on employing monoethanol amine (MEA) as a scrubbing reagent. MEA is a mildly alkaline substance which reacts with the principal acid gases, CO_2 and H_2S, to form the MEA bicarbonate and MEA hydrosulfide which are soluble in the solution. The pregnant solution can then be pumped to a regenerating vessel where the CO_2 and H_2S can be stripped by steam heat. The CO_2, H_2S, and H_2O effluent gas can be further processed to recover the H_2S. The regenerated MEA solution is then recycled to the absorber. The MEA processes have been highly developed, and by staging and optimization of conditions, H_2S and CO_2 may be reduced to very low levels. The principal disadvantage of this process for desulfurization of coal gas is the fact that the MEA reacts irreversibly with COS which is present in fuel gases to a low but nonetheless significant extent. The MEA-COS reaction product is not regenerable, so significant losses of MEA result. Other amines may be used, such as diethanolamine (DEA), which are less susceptible to degradation by COS, but these are generally more expensive and not as effective in removing H_2S.

Hot-Potassium Carbonate Scrubbing Processes

The Bureau of Mines developed the hot-potassium carbonate scrubbing process for the removal of CO_2 from coal gas to upgrade its heating value. It was discovered that H_2S was also removed, and that the $KHCO_3$ and KHS, which are formed in the solution by the reaction of the CO_2 and H_2S with the K_2CO_3, could be decomposed to regenerate K_2CO_3 which would then be recycled to the absorber. The process operates in principle quite similarly to the MEA process, but has the advantage that it can absorb the CO_2 and H_2S at somewhat more elevated temperatures (i.e., 230 F) than would be possible with MEA; also, the K_2CO_3 is not as volatile as MEA. The hot-potassium carbonate process works best at about 300 psig, which is favorable for use with high-pressure coal gasification processes. A large fraction of the required steam for stripping the H_2S and CO_2 in the regeneration step comes from flashing of water vapor in the reduction of the pressure from 300 psig to about 20 psig during transfer of pregnant solution from the absorber to the regenerator.

A particular advantage of the hot-carbonate scrubbing process is that the COS which is present in coal gases does not affect the absorbent solution as in MEA. Instead, COS hydrolyzes in the solution (Ref. 54) according to the reaction:

$$COS + H_2O \rightarrow CO_2 + H_2S \qquad (5)$$

The resulting H_2S and CO_2 would then react with the K_2CO_3 in the solution.

A disadvantage of the hot-carbonate scrubbing system is the rather limited carrying capacity of the solution, being about 3 ft^3 of acid gases per gallon of absorbent solution (Refs. 54 and 55). This requires that a very large amount of potassium carbonate solution be recycled between the absorber and regenerator. For example, a system to desulfurize coal gas for a 1000-Mw first-generation power plant would process about 2450 lb moles of gas per minute and require a circulating liquor flow of 51,580 gpm in the absorbers. This liquor flow determines the size of the absorber and stripper vessels, their numbers, and, consequently, the capital investment in the plant.

Steam for stripping the loaded solution in the regenerators has been estimated to be approximately 1 lb of steam per 6 ft^3 of acid gas. This value is well above the estimate of 1 lb of steam per 95 ft^3 of acid gas indicated in the original pilot work. A detailed flow sheet for a hot-potassium carbonate absorption system is given in Fig. 3.

Sulfinol

The Sulfinol process, developed by Shell (Refs. 56 and 57), uses a mixture of an organic solvent, sulfolane (tetrahydrothiophene dioxide), and an alkanolamine, i.e., di-isopropanolamine, as chemical reactant, in a conventional scrubbing system. The organic solvent provides high capacity by physical solution, and the amine provides low H_2S concentration in the gas by chemical reaction. Thus, the combined solution has a good acid-gas capacity over a broad range of partial pressures. Steam requirements for regeneration are low since part of the acid gas is physically held. Because of the high capacity, less circulating solution is required. The solution has a low heat capacity, and the equilibrium for physical solution is sensitive to temperature. These factors also contribute to low steam requirements for regeneration. The solvent absorbs COS, as well as mercaptans and hydrocarbons. The disadvantages are that the solvent is very expensive and that hydrocarbon absorption complicates sulfur recovery in a Claus unit. However, hydrocarbon absorption is significant only for aromatic hydrocarbons and high-molecular-weight aliphatic hydrocarbons, so that the absorption may not necessarily be a problem, depending on the composition of the gas to be treated (Refs. 58 through 61).

A flow sheet for a Sulfinol absorption system for use in conjunction with the partial oxidation of residual oil is contained in Section III.

The following conditions were assumed:

Absorber Temp., F	110	
Pressure, psia	300	
Stripper Temp., F	125	
Pressure, psia	30	
Gas Composition %	In	Out
CO	2.09	0.85
H_2S + COS	0.72	0.91
Solution Capacity	4 ft^3 acid gas/gal	
Solution Loss - (includes entrainment, leakage, vapor loss, and degradation)	2 lb/million ft^3 acid gas	

Seaboard Process

This was one of the earliest commercial gas scrubbing methods, having been introduced industrially by the Koppers Co. in 1920. The sour gas is treated with a dilute sodium solution (3.0 to 3.5% aqueous). The H_2S, CO_2, and HCN (if present) react with the absorbent to form sodium hydrosulfide and bicarbonate. The pregnant solution is pumped to a separate vessel where the sodium carbonate is regenerated by blowing with air. There is no recovery of the H_2S and CO_2 gases. This process must now be considered obsolete.

Vacuum Carbonate

This process was an outgrowth of the old Seaboard process and was developed by Koppers Co. to provide a means of recovering the H_2S from the rich solution. Steam is used to strip the H_2S from the solution; the steam bearing H_2S and CO_2 is then condensed, and the H_2S and CO_2 gases are further treated to reclaim the H_2S.

A vacuum of 25 in. Hg is employed to minimize the steam requirements. A very large circulating liquor rate must be provided for in a large-scale operation because of the much lower solubility of sodium carbonate in water compared with potassium carbonate.

The Phosphate Process

This process, developed by Shell Development Co., is similar to the hot-potassium carbonate process, except that tri-potassium phosphate is used as the absorbing solute. The chemical reactions which occur are similar, since tri-potassium phosphate solutions are alkaline. For example,

$$K_3PO_4 + H_2S \rightarrow KHS + K_2HPO_4 \tag{6}$$

Tri-potassium phosphate is quite soluble in water, so that concentrated solutions may be employed which is an advantage over the vacuum carbonate process. The H_2S may be recovered by steam stripping. A disadvantage is that somewhat more energy in the form of the stripping steam is required than in other scrubbing systems such as the MEA processes.

Wet Iron Box Process

A classical method of desulfurizing gas is the Wet Iron Box process (Ref. 52) which has been in use, particularly in England, since the middle of the 19th century. Modern modifications which are still in use are the Ferrox and Manchester processes. In this type of process, the H_2S in the sour gas reacts with moist Fe_2O_3 or $Fe(OH)_3$ in a sodium carbonate slurry to form $Fe_2S_3 \cdot H_2O$. The pregnant solution or slurry is then pumped to a regenerator vessel where it is contacted with air for a period of time. The regeneration step consists of the oxidation of the ferric sulfide to form elemental sulfur and ferric oxide. The sulfur floats as a froth on the solution and is removed either by filtration or in a weir box. The sulfur recovered in the Ferrox method is usually quite impure and is usually not further processed for economic sulfur recovery. In the modification known as the Manchester Process, the sulfur can be obtained in 90 to 95% pure form which may be sufficient for further economic sulfur recovery.

The Thylox Process

This process uses as the absorbent a sodium or ammonium arsenate solution which reacts with the H_2S in the sour gas to form a thioarsenate plus elemental sulfur. The sulfur is removed from the solution by filtration or flotation, and the solution is regenerated by blowing with air. The Thylox process is now considered to be obsolete. A modern modification of it is the Giammarco-Vetrocoke process, which uses a sophisticated control over the reaction steps by pH and concentration monitoring. The sulfur recovered by this process has been claimed to be pure enough for agricultural purposes. The Giammarco-Vetrocoke process is suitable only for relatively small plants with sulfur outputs of about 25 tons per day (Ref. 63). This process has not been industrialized in the USA.

Dry Absorption Systems

Hot-Ferric Oxide Process

This process (Ref. 64), also referred to as the Appleby-Frodingham process, is a dry process. The sour gas is reacted with hot ferric oxide powder or pellets at about 600 F in a fluidized bed. About four stages of absorption would be needed to reduce the H_2S and COS content of the sour gas to meet third-generation sulfur levels. The following reactions evidently take place:

$$Fe_2O_3 + 3H_2S \rightarrow FeS \cdot FeS_2 + 3H_2O \qquad (7)$$

$$Fe_2O_3 + 3COS \rightarrow FeS \cdot FeS_2 + 3\ CO_2 \qquad (8)$$

The solid absorbent is contacted with the gas countercurrently. Plant data indicate that 98% of the H_2S in fuel gas can be removed by reaction. No data are available on COS removal levels. However, the equilibrium for COS removal is favorable, and therefore, 98% COS removal was also assumed.

The pregnant solids leave at the bottom of the absorber and are conveyed to the regenerator where the ferric oxide is regenerated by an air blast, thus:

$$FeS \cdot FeS_2 + 9/2\ O_2 \rightarrow F_2O_3 + 3SO_2 \qquad (9)$$

It is possible to balance the solid absorber recycle rate and the air blast rate so that the exit gas from the regenerator is at a concentration of SO_2 (about 7%) and temperature suitable for direct feed to a standard contact sulfuric acid plant. The scrubbed gas leaves at the top of the absorber with virtually no loss of sensible heat. These features make the hot ferric oxide process quite interesting in any clean fuel process where the clean product gas is required to be hot (600 to 650 F).

The disadvantage of the process is that the sulfur is recovered in the form of concentrated sulfuric acid which requires expensive storage tanks and the inevitable inventory control problems. There exists the possibility that the exit gas bearing the SO_2 could be combined with a stream of raw gas to provide a feed suitable for recovery of sulfur in a Claus unit.

A flow sheet for the hot ferric oxide process used in conjunction with a partial oxidation of coal system is shown in Fig. 4.

Dolomite Acceptor Process

The use of calcined dolomite or limestone as a high-temperature H_2S absorbent has been proposed by several sources, e.g., Refs. 22 and 23. Calcined dolomite ($CaO \cdot MgO$) reacts with hydrogen sulfide according to the following reaction:

$$CaO \cdot MgO + H_2S \rightarrow CaS \cdot MgO + H_2O \tag{10}$$

The partial pressure of hydrogen sulfide over calcined dolomite for gases which would be at the conditions typical of advanced gasification process (22 atm pressure, 5.9% H_2O) is shown in Fig. 5. The initial hydrogen sulfide partial pressure in the gasifier off-gas would be 0.134 atm. It is shown in Fig. 5 that 95% of the H_2S in the gasifier off-gas could be removed at 2000 F, 98.5% at 1600 F, and 99.2% at 1450 F. Since the lowest possible residual sulfur would be the goal for advanced gasification processes, desulfurizer operation at 1450 F was assumed. This temperature level would be close to the lowest temperature where reasonable rates for reaction (10) could be maintained. At 1450 F the absorption of COS on calcined dolomite

$$CaO \cdot MgO + COS \rightarrow CaS \cdot MgO + CO_2 \tag{11}$$

is also favorable; thus, it was assumed that 99% approach to equilibrium could be achieved with respect to overall sulfur removal.

In addition to the absorption of hydrogen sulfide, calcined dolomite absorbs CO_2 at 1450 F.

$$CaO \cdot MgO + CO_2 \rightarrow CaCO_3 \tag{12}$$

The partial pressure of CO_2 in the off-gas from an advanced gasifier would be 0.616 atm, while the equilibrium partial pressure at 1450 F would be 0.2 atm. It was assumed that a 90% approach to equilibrium could be obtained in the desulfurizer (Ref. 23). Since the reactions which would occur in the desulfurizer would be exothermic, the gasifier off-gas must be cooled to 1150 F in order to maintain a temperature of 1450 F in the desulfurizer.

The regeneration of spent acceptor is accomplished by reaction with steam and carbon dioxide.

$$CaS \cdot MgO + CO_2 + H_2O \rightarrow CaCO_3 \cdot MgO + H_2S \qquad (13)$$

The equilibrium concentration of H_2S in the dry gas after regeneration at 22 atm using an equimolar feed of steam and CO_2 is shown in Fig. 6. Temperatures below 1175 F would be required to obtain an off-gas with a hydrogen sulfide concentration greater than 20%. For the dolomite process regeneration typically could be assumed to take place at 110 F, and H_2S concentrations could be assumed to approach 90% of its equilibrium value. A temperature of 1100 F would be maintained in the reactor by raising the steam in an internal coil. The steam would be subsequently used for reaction (11). At 1100 F the partial pressure of CO_2 over MgO would be greater than 50 atm. Therefore, the magnesium oxide would not participate in the recarbonation.

After regeneration the dolomite must be calcined before recycle to the desulfurizer. This is accomplished by reversing reaction (12). The equilibrium partial pressure of CO_2 over calcined dolomite is approximate 5 atm at 1900 F. To raise the temperature of the regenerated dolomite from 1100 to 1900 F and provide heat for the endothermic calcination, a portion of the clean fuel gas would be combusted.

Fluidized-bed reactors would be used for the desulfurizer, regenerator, and calciner. Also, a hot carbonate scrubbing system could be used for the recovery of CO_2 from the regenerator off-gas for eventual recycle to the regenerator.

A flow sheet for this process is included in the discussion of third-generation fuel processes in Section IV (Fig. 20).

Claus Systems for Recovery of Elemental Sulfur

The desulfurization of fossil fuels usually requires some plan for the disposition of the sulfur which is removed. The Seaboard process discussed above successfully removed the H_2S from the sour gas, but there was no provision for the recovery of the H_2S which was discharged to the atmosphere after stripping of the absorbing solution. This arrangement is, of course, no longer acceptable, and various schemes have been developed to recover the sulfur in a form that has economic value if possible. The most important of these schemes which involve selective oxidation of H_2S to elemental sulfur have been classified together as Claus systems, named after the inventor of the original selective oxidation process which was invented about 1890. An excellent review of the original Claus process and three modifications of it are contained in Ref. 65. The following discussion is based on that review, plus additional information supplied by the Ralph M. Parsons Co., which firm has made a specialty of the design and construction of Claus plants (Ref. 66). Additional valuable information was supplied by the C&I Girdler Corp., Gas Processing Division, Cincinnati, Ohio, which also designs and constructs Claus plants (Ref. 67).

The original Claus scheme was based on the oxidation of H_2S in a controlled quantity of air, thus:

$$H_2S + \tfrac{1}{2}O_2 \xrightarrow[500 \text{ to } 600 \text{ F}]{\text{catalyst}} H_2O + S \tag{13}$$

The catalyst can be a variety of solid substances, but the most important and simplest is bauxite. The principal difficulty with the original Claus reaction is control of the temperature from this exothermic reaction (-48 to -58 kcal/g-mole H_2S). This was usually done by restricting the space velocity of the reactants in the catalytic combustion chamber. The product gases must be cooled below the dew point of the sulfur vapor in order to collect the product.

An important modification is the I.G. Farbenindustrie process which involves splitting the sour gas stream into one-third and two-thirds by volume and oxidizing thus:

$$H_2S + 3/2\ O_2 \rightarrow H_2O + SO \quad \text{(one-third)} \tag{14}$$

$$2H_2S + SO_2 \xrightarrow{\text{catalyst}} 2H_2O + \frac{3}{x}\ S_x \quad \text{(two-thirds)} \tag{15}$$

The first reaction is highly exothermic and is carried out in a waste heat boiler to recover energy as steam to be used elsewhere in the process. The second reaction is only mildly exothermic and is carried out in the catalytic combustion chamber with good control of temperature and a higher permissible space velocity.

For gases containing H_2S in relatively dilute form or in relatively pure form, further modifications were required to achieve higher recoveries of the product sulfur and to control the process. These modifications involve staging the H_2S-SO_2 reactions with interstaging cooling and sophisticated recycling of the gas streams to regulate the temperature and equilibrium concentration of the gas constituents. These methods, called the Great Lakes Carbon Co. process and the Jefferson Lake processes, are discussed in Ref. 65. A flow sheet of one of the latter processes (Fig. 7) was prepared for use in the process design for Claus plants used in this study.

Because the expected composition of the gas emitted from the stripping column of the hot carbonate scrubbing system would be dilute in H_2S and rich in CO_2, a recycle of sulfur with combustion to SO_2 was considered to be necessary (Ref. 66). The SO_2 stream would then be mixed with the raw gas in a two-stage catalytic converter and the product sulfur washed out in a tower which would be scrubbed with molten sulfur. The expected recovery of the sulfur would be 90%. This process is similar to one of the Jefferson Lake modifications discussed in Ref. 65.

SELECTION OF FUEL DESULFURIZATION AND SULFUR RECOVERY PROCESSES

The review and discussions presented in Section I allowed the early elimination of several methods of desulfurization, e.g., physical methods such as flotation and solvation of coal, and advanced hydrodesulfurization of residual oil. The processes remaining all involve the gasification of the feedstock, be it coal or oil. These processes, which are described in detail in subsequent sections, are summarized in Table 4 for coal and in Table 5 for oil. In order to fully evaluate a fuel process it must be considered in combination with the advanced-cycle power system that it would fuel, since the size, and thus the cost, of a given fuel process is a function of power system economics.

The primary basis of comparison in Tables 4 and 5 is the estimated fuel cost. Since all the processes listed surpass or meet the sulfur emission goals for the particular time period, this criterion becomes secondary. It should be noted, however, that liquid scrubbing with hot potassium carbonate offers the lowest sulfur emissions.

Section II - First-Generation Fuel Desulfurization Process

This section contains detailed process descriptions, material balances, capital cost estimates, and annual owning and operating cost estimates for those coal and residual oil desulfurization processes identified in the preceding section as offering the greatest potential for providing low-cost desulfurized fuel to first-generation power systems. Desulfurization processes analyzed include coal gasification processes for base-load gas turbine and COGAS power systems and residual oil hydrodesulfurization for conventional steam power systems.

FIRST-GENERATION COAL DESULFURIZATION

The coal conversion processes identified in the preceding section would involve gasification of coal followed by multiple scrubbing systems to remove dust, tar, and sulfur compounds. A process based on the use of Lurgi-type, dry-ash, fixed-bed gasifiers was chosen for detailed analysis because these gasifiers are the only presently available commercial gasifiers operating at elevated pressures. Detailed material balances and cost estimates were prepared for high-pressure processes designed for base-load gas turbine and COGAS power systems. In addition, a less detailed analysis was made of a fuel process incorporating low-pressure gasification to determine the effect of gasification pressure level on desulfurized fuel cost for a COGAS power system.

High-Pressure Gasification Process for COGAS Power System

As previously noted, the high-pressure coal gasification process was based on the use of Lurgi gasifiers because they are the only commercially available high-pressure gasifiers. These gasifiers were first developed in 1933, and twelve gasification plants have since been built to produce town gas and synthesis gas. These plants are, for the most part, operated with oxygen, but a new installation designed to operate with air is now being constructed (Ref. 68) to provide fuel gas for the Kellermann power station in Lunen, West Germany. The Kellerman installation will not incorporate any scheme for desulfurization of the fuel gas, whereas the gasification process described below would incorporate a hot-carbonate sulfur scrubbing system.

Process Description

A flow sheet of the high-pressure gasification process to produce desulfurized fuel for the first-generation COGAS power system is depicted in Fig. 8. This process consists of five major sections: coal preparation, gasification, waste heat removal plus tar and dust recycle, sulfur removal, and by-product sulfur recovery.

Coal Preparation

Coal would be transported from a 90-day storage pile at the rate of 9414 tons per day (including 13% moisture content) by means of an underfeed system of three parallel conveyors. These conveyors would feed a continuous belt conveyor which in turn would feed two primary jaw crushers operating in parallel. The crushers would reduce the coal to a 1-inch consist. Coal would then be screened to an acceptable consist for the Lurgi gasifiers, 28-mesh x 1 in. It is estimated that approximately 19% of the as-mined coal, or approximately 1800 tons/day, would be less than 28-mesh size. This undersize coal fraction would be transported to six briquetting machines operating in parallel. Here the size consist of the fines would be increased to that acceptable for the Lurgi gasifiers. Tar produced during gasification could be used as a binder for the briquettes. Coal from the crushers and briquetting machines would be transported by an inclined belt conveyor system to a belt conveyor distributing system feeding each gasifier.

Gasification

Gasification would take place continually in a battery of 23 12-ft-diameter Lurgi dry-ash gasifiers. Each gasifier consists of a coal hopper mounted above a coal lock hopper which is, in turn, mounted above the cylindrical gasifier. The gasifier is equipped with a mechanical coal distributor and rotating grate. Mounted below the gasifier is an ash lock hopper. Coal fed into the gasifier through the lock hopper descends by gravity to the grate. Air and steam are introduced into the gasifier below the grate and travel upward to the direction of coal flow.

The gasification reactions proceed in three distinct zones within the fixed bed of coal in the gasifier. Directly above the grate, coal is combusted in air to form CO_2 according to the following reaction.

$$2C + O_2 \rightarrow 2CO \qquad (16)$$

Temperatures in this zone are kept below the fusion temperature of the ash by simultaneous, endothermic reaction of steam with carbon (Eq. (1)). The combustion gas then flows upward through the bed into the second zone. Here, further endothermic gasification reactions occur along with the water-gas shift reaction.

$$C + CO_2 \rightarrow 2CO \qquad (17)$$

$$C + 2H_2 \rightarrow CH_4 \qquad (18)$$

$$CO + H_2O \rightarrow CO_2 + H_2 \qquad (4)$$

In this second zone, devolatilization of the coal occurs simultaneously with gasification yielding additional methane and higher-order hydrocarbons. The gases then flow upward into the third zone near the top of the bed. Here, drying and additional devolatilization of the coal occur producing higher-order hydrocarbons and tar. Fuel gas finally exits the reactor at approximately 950 F. Ash would drop through the rotating grate and preheat air and steam feeds. Ash would then be discharged from the gasifier through a lock hopper to a hydraulic sluicing system for treatment and subsequent disposal.

Air required for gasification, approximately 360,000 ft^3/min, would be supplied by two 50,000-hp, 3-stage, centrifugal compressors operating in parallel. Electric drives were chosen for the air compressors because of the high efficiency of electrical generation in the COGAS system. In a first-generation COGAS system, fuel could be converted to electricity at 47% efficiency, whereas steam turbines would operate in cycles with 40% or less thermal efficiency. Since capital costs for electric motors and steam turbines are comparable, electric drives were selected for this process and other processes in which insufficient steam could be raised by waste heat boilers in the gasification section of the plant to drive steam turbines.

Approximately 470,000 lb/hr of 325 psi/450 F steam would be required for gasification, well beyond that raised in the cooled walls of the gasifier. Approximately 60% of this steam would be raised in waste heat boilers. The remaining 40%, approximately 180,000 lb/hr, would be raised in a gas-fired boiler using approximately 4% of the clean fuel gas made in the process. The use of steam raised in the power cycle of the plant was considered as an alternate source for the steam. However, since some of the advanced power systems investigated could not raise the required amount of steam to drive the compressors, this alternative was not investigated in detail.

Waste-Heat Recovery

Gas exiting the gasifiers at 950 F would be cooled to 550 F in waste heat boilers. This is similar to the system used at SASOL (Ref. 69). About 60% of the steam required for gasification would be raised in these boilers. Some tar would be expected to be condensed on the tubes of the waste-heat boiler. At the estimated tube temperature of 450 F, tar would be expected to be free-flowing and relatively stable. To take oil fouling into account, an overall heat transfer coefficient of 15 Btu/hr/ft^2 F was assumed. Each gasifier would have its own waste-heat boiler.

The use of a scrubbing tower employing hot tar as a scrubbing agent was investigated as an alternative to waste-heat boilers. Gas could be cooled by countercurrent contact with hot tar, and the hot tar could then be cooled in waste-heat boilers to produce steam. In such a system the amount of tar absorbed would be small compared to the tar circulation rate. Therefore, the tar would have a long residence time in the system at high temperatures and would probably become unstable. Also, capital costs appear greater than for a simple waste-heat boiler; thus, this option was discarded.

Residual tar and dust would be removed from the gas by water scrubbing. The scrubber would also reduce the gas temperature to 290 F. Tar would be separated from the water by gravity, combined with tar condensed in the waste-heat boiler, and either recycled to the gasifier or used as a binder in briquetting. Water would be recycled to the scrubber. Each gasifier would have its own scrubber.

Off-gases from the water scrubbers would then be further cooled to 250 F in water-cooled heat exchangers. Six parallel trains of heat exchangers would be required. After cooling, the actual gas volume flow would be approximately 47,000 ft^3/min at 250 F, 317 psia.

Sulfur Removal

Sulfur compounds (0.57% H_2S and 0.06% COS) would be removed from the cooled, tar- and ash-free fuel gas by a hot-carbonate scrubbing system (Ref. 79). The cooled fuel gas would pass through to a battery of six 12-ft-diameter absorption columns operating in parallel. Here, the fuel gas would be contacted countercurrently with a 30% solution of potassium carbonate which would absorb the COS, H_2S, and CO_2. Based on experimental data (Ref. 53), it appears that it would be possible to remove 85% of the H_2S and COS present in the fuel gas while removing approximately 12% of the CO_2. In doing this, the sulfur content of the fuel gas would be reduced to 221 g/million Btu, well below the first-generation goal of 450 g/million Btu. The desulfurized fuel gas would leave the absorption tower and pass through a high-efficiency entrainment separator, not shown in Fig. 11, to recycle the potassium carbonate liquor. The fuel gas would then be delivered to the power station at 230 F and 315 psia.

The spent potassium carbonate liquor, laden with H_2S, COS, and CO_2, would be pumped to a battery of six 12-foot-diameter stripping towers operating in parallel in which the pressure would be dropped from 315 psia to about 21 psia and the carbonate liquor would be stripped with steam. The regenerated carbonate solution would then be recycled to the absorber. The gas effluent from the strippers would contain approximately 10.9% H_2O, 67.5% CO_2, 21.6% H_2S (by volume), and minor amounts of inert gases. The gas would then be cooled to 140 F in water-cooled heat exchangers and passed to the Claus sulfur recovery system.

A standard Claus recovery system would be used to recover by-product sulfur from the stripped acid gases. In this system, approximately one-third of the H_2S in the gas would be combusted to produce SO_2. The effluent gas would then be cooled and pass through two bauxite catalyst beds, operating, in series, where reaction (15) would occur. The gases would be cooled again to condense the sulfur. Heat recovered during these two cooling processes would be used to raise some of the process steam required to regenerate the carbonate liquor. The sulfur recovery system is estimated to be about 90% efficient. Sulfur compounds not removed in the Claus system would be incinerated and discharged through a stack.

Although the first-generation process described herein was designed for 85% removal of the sulfur compounds, higher removal rates would be possible with the hot-carbonate system. By using higher columns and split stream techniques, more than 99% sulfur removal could be effected, resulting in a net sulfur content in the fuel below 15 g/million Btu. However, at high H_2S removals, high CO_2 removals would also be obtained. Since there would be about 20 times more CO_2 in the gas than H_2S, the stripped acid gas would be expected to contain about 5% H_2S, far below what would be required for a Claus converter (approximately 20%). To accommodate these low concentrations, expensive modifications in the Claus process, such as sulfur recycle schemes, could be employed. In addition, at high H_2S and CO_2 removals, steam requirements for hot-carbonate stripping would increase. Therefore, the 85% sulfur removal level was considered to be optimum for the first generation because capital and operating costs would be reasonably low and final sulfur emissions would be well below the prescribed goal for a first-generation plant. Similarly, if higher sulfur removal rates from the Claus system were desired, the off-gas from the Claus system potassium carbonate scrubbers could be recycled to the scrubbing system. This modification was not deemed necessary to meet first-generation sulfur removal goals.

Material Balance

Raw material requirements and process yields are summarized in Table 6. For a nominal 1000-Mw plant with a 70% operating factor, approximately 2.4 million tons/year of coal would be required. Additionally, 82 Mw of power would be consumed, mainly to supply compressed air for the gasifiers. Therefore, the 1000-Mw integrated power station would have a net power output of 918 Mw. Clean fuel gas in the amount of 686,622 ft^3/min (based on standard conditions of 60 F, 14.7 psia) would be supplied to the power generating system at 230 F and 315 psia. The gas would have a higher heating value of 172.7 Btu/ft^3 and a sensible heat above 60 F of 3.3 Btu/ft^3. The sulfur content of the gas would be 221 g/million Btu (based on HHV plus sensible heat). In addition, 52,067 long tons/year of by-product sulfur would be produced.

Capital Cost

A capital cost estimate for the process is given in Table 7. Total fixed capital was estimated to be \$44.5 million corresponding to \$48/net kw. The major capital cost areas were: gasification - 45%, air compression - 20%, coal handling and preparation (mainly briquetting) - 11%, and general facilities - 9%. Additional items make up approximately 15% of the capital cost.

Recent information from the Lurgi Company (Ref. 71) indicates that 16-ft-diameter gasifiers are being contemplated, and that the use of these units could reduce the capital cost of the gasification from approximately \$20 million to \$15 million. This would result in an 11% reduction in total capital costs.

Annual Owning and Operating Costs

Annual owning and operating costs for the Lurgi dry-ash gasification process are also presented in Table 7. Operating costs are based on a power station producing 918 Mw of net electrical power with a 70% operating factor. Maintenance was estimated at 5% of fixed capital. Labor requirements were estimated at 38 men/shift at \$15,000/man/shift, including payroll and accounting costs, overhead, and supervision. Utility costs were estimated as follows: cooling water @ 2¢/1000 gal, process water @ 30¢/1000 gal, and steam @ 50¢/1000 lb/hr. Sulfur was credited at \$25/long ton. If no credit were taken for sulfur, operating costs would increase by 0.23 mills/kwhr, or roughly 10% of the total operating cost (exclusive of fuel) attributable to the gasification plant. Coal requirements are 0.85 lb/kwhr, and coal costs were evaluated at 20¢/million Btu (\$4.42/ton). The overall system efficiency, net kw/coal in, for the first-generation gasification process is 36.1%. Capital-related costs, including depreciation, taxes, insurance, and interest, were estimated at 14% of fixed capital cost following utility financing practices. The total annual owning and operating costs attributable to the gasification plant were estimated to be approximately \$22.4 million/yr corresponding to a clean fuel cost of 3.98 mills/kwhr. A decrease in capital costs of \$5 million, the possible savings which could be accrued by using larger-diameter Lurgi gasifiers, would result in a 0.17 mills/kwhr, or 4%, reduction in operating cost.

The equivalent fuel cost of 3.98 mills/kwhr can also be interpreted in terms of cost per million Btu. For the first-generation COGAS plant, this cost would be approximately 43¢/million Btu, a figure slightly more than double the cost of the raw fuel.

High-Pressure Gasification Process for Base-Load Gas Turbine Power System

A cost estimate was also prepared for a high-pressure process based on the use of Lurgi gasifiers with hot-carbonate scrubbing to supply clean fuel to a first-generation, base-load gas turbine plant. Fuel requirements for this gas turbine plant would be 161.4 million Btu/min of clean gas at 315 psia. To supply these requirements, the Lurgi gasification system would have to process 3.211 million tons/yr of coal and produce 916,778 ft^3/min of fuel gas. This gas would have the same composition as given in Table 6. The process would require 109 Mw of electrical power.

A material balance for this system is given in Table 8, and the capital and annual owning and operating costs are summarized in Table 9. The clean fuel cost would be 5.21 mills/kwhr, corresponding to an equivalent fuel cost of 40.4¢/million Btu.

Atmospheric-Pressure Gasification Process for COGAS Power System

Costs for an atmospheric-pressure, fixed-bed gasification process were also developed to serve as a comparison with a fixed-bed, high-pressure process. A flow sheet of this process using Wellman-Galusha gasifiers (Refs. 17 and 28) is shown in Fig. 12. This process was sized to provide fuel to a 1000-Mw nominal COGAS power system. Schematic flow sheets for the high- and low-pressure processes are similar except for the location of the main compressors. In the high-pressure process the main compressor would be located before the gasifier, whereas in the low-pressure process the main compressor would be located before the hot-carbonate scrubbing system. Since the gas stream at this point would contain the partial oxidation products and be at somewhat elevated temperature, the volume flow rate of gas would be approximately twice as much as in the high-pressure process, requiring increased compressor power.

Process Description

Wellman-Galusha gasifiers operate on a more restrictive coal size consist than the Lurgi gasifiers, $\frac{1}{2}$ inch x 2 inch. Therefore, a greater percentage of the coal must be briquetted prior to gasification. Steam requirements for gasification could be supplied by saturating air in the jacket of the gasifier. Off-gases would exit the gasifiers at a higher temperature than the Lurgi gasifiers (1250 F versus 950 F). As a result, approximately 575,000 lb/hr of 150 psia/450 F steam could be raised in waste-heat boilers. About 20% of this steam would be used to meet the steam requirements of the acid gas stripping columns. The remainder would be used in steam turbines to supply about 20% of the power requirements of the gasification system. After cooling and scrubbing to remove tar and dust, the fuel gas would be compressed to about 21 atm and scrubbed in a hot-carbonate acid gas scrubbing system. Since there would be less CO_2 in the off-gas from a Wellman-Galusha gasifier than from a Lurgi gasifier (3% versus 13%), a greater amount of H_2S would be removed while maintaining a suitable concentration of H_2S for feeding to a Claus converter. About 92% of the H_2S would be removed from the fuel gas. The stripped acid gas would contain approximately 30% H_2S.

Material Balance

A summary of raw material requirements and process yields for the low-pressure gasification process is shown in Table 10. Coal in the amount of 2.47 million tons/yr would be required for a nominal 1000-Mw COGAS plant system. Electrical requirements would be 140 Mw, resulting in a net electrical production of 860 megawatts from the integrated power station. Clean producer gas in the amount of 743,330 ft^3/min would be supplied at 230 F and 214 psia. The gas would have a higher heating value of 160.4 Btu/ft^3 and a sensible heat above 60 F of 3.2 Btu/ft^3. The sulfur content of the gas would be 117 g/million Btu, or approximately 25% of the first-generation sulfur standard. Approximately 57,000 long tons/year of by-product sulfur would be produced.

Capital Cost

Capital cost estimates for the process are shown in Table 11. Total fixed capital was estimated to be $71.4 million corresponding to $82/net kw. Total investment was estimated to be $73.5 million. Major cost items were gasification, 45% of fixed capital, fuel gas compression 28%, and coal crushing and briquetting 8%. Neglecting the cost of gasification itself, fixed capital costs for the low-pressure, fixed-bed process were roughly $15 million higher than those for the fixed-bed, high-pressure process. This demonstrates the advantages of high-pressure gasification for use in conjunction with an advanced-cycle power system.

Annual Owning and Operating Costs

Annual owning and operating costs were based on a plant producing 860 Mw of net power at a 70% operating factor. These costs are also summarized in Table 11. The total fuel processing cost for this system is estimated to be 4.80 mills/kwhr, which is approximately 20% higher than the fuel cost for the high-pressure gasification system. The overall system efficiency of the low-pressure process would be 33.2%, compared to 36.1% for the high-pressure process.

FIRST-GENERATION OIL DESULFURIZATION

The promising residual oil desulfurization processes identified in Section I involve hydrodesulfurization of selected oil fractions followed by reblending with selected remaining fractions. The process selected for detailed performance and cost analysis involved hydrodesulfurization of vacuum gas oil followed by reblending with the vacuum resid. Cost and performance estimated for other processes involving hydrodesulfurization are also presented. These oil desulfurization processes would not be effective in removing all of the metals originally contained in the residual oil. These metals, in combination with the sulfur remaining in the fuel oil, would produce unacceptable rates of corrosion in advanced power systems utilizing gas turbines or high-temperature fluid heaters. Consequently, this fuel was considered for use only in relatively low-temperature, externally fired power systems such as conventional steam systems.

Vacuum Gas Oil Hydrodesulfurizations

Process Description

A flow sheet for the vacuum gas oil (VGO) hydrodesulfurization process is shown in Fig. 13. For this process, long resid would be preheated by the reactor effluent and would then enter a direct-fired furnace to be heated to 700 F prior to entering the vacuum still (10 mm Hg, maintained by two-stage steam ejector). The VGO from the still would be mixed with recycle and make-up hydrogen, would be heated to 680 F in a direct-fired heater, and then would enter the hydrodesulfurization reactor. The hydrogen make-up was based on a hydrogen consumption of 2600 ft^3/bbl or 65 ft^3/lb of sulfur removed. These figures include chemical consumption, as well as losses. Make-up hydrogen would be produced on site by steam reforming of natural gas. The reactor would operate at 800 psia, corresponding to a hydrogen partial pressure of about 625 psia. The liquid hourly space velocity would be set at 1.92 hr^{-1}. These conditions of pressure, temperature, and space velocity would give a sulfur removal of 78% of the VGO, which represents 60% of the long resid. Catalyst life has been estimated as three years with yearly regeneration. This corresponds to about 65 bbl of oil per pound of catalyst.

The effluent from the reactor would be used to preheat the feed and would then be further cooled in a steam generator and water heat exchanger. A high-pressure separator would be used to separate hydrogen for recycle, since most of the hydrogen sulfide would be dissolved in the VGO. A low-pressure separator would remove most of the hydrogen sulfide, as well as some hydrogen and light hydrocarbons. The oil from the low-pressure separator would undergo heat exchange with the fractionater bottoms, then would be further heated in a direct-fired heater to 620 F and fractionated into a cracked gasoline and naptha fraction, a middle distillate, and a heavy gas oil. After cooling, the products would be pumped to storage tanks, and the desulfurized heavy gas oil would be blended back with the vacuum resid to provide the fuel oil for power generation. A portion of this fuel would be used for the process in the direct-fired heaters. The tail gas from the fractionator would be mixed with the off-gas from the low-pressure separator. The H_2S in the gas mixture would be removed in a Sulfinol absorption system. The gas, now free of H_2S, would be used as fuel for the furnace preheaters, and the H_2S would be stripped from the absorber solution and converted to sulfur in a two-stage, direct-flow Claus system. The unconverted H_2S (about 5%) from the Claus system would then be incinerated and discharged through a stack.

The sulfur emitted from the incinerator stack and the sulfur released from the combustion of the fuel oil were considered to be part of the total power station sulfur emissions. Therefore, the percent sulfur in the product fuel oil would be reduced to about 1.65%, instead of about 1.8% which would otherwise be required to meet first-generation requirements. This is also near the practical maximum limit of sulfur removal that can be achieved with a vacuum gas oil hydrodesulfurization.

Material Balances

A nominal material balance corresponding to a plant that would produce enough desulfurized fuel for one 1000-Mw conventional steam power system is given in Table 12. The stream numbers indicated in this table correspond to those identified in Fig. 13. In determining this nominal material balance it was assumed that a portion of the product oil would be used for fuel to run the process, that the product oil would be cooled for storage, and that credit for middle distillate would be taken. The material balance presented in this table was used to size and cost the equipment. Actual material balances for first-generation processes would differ slightly from that shown in the table, depending on whether natural gas were used to fuel the process instead of product oil, or whether credit were taken for middle distillate or the distillate were reblended into the product oil, and whether the oil were burned hot or cold in the power system. These differences are discussed in the following paragraphs.

In addition to sizing a fuel plant for a single 1000-Mw power system, a larger plant sized to provide fuel to five 1000-Mw power stations may also be considered to take advantage of economies of scale. These plants were assumed to be located in the Northeast. In addition, a large plant was assumed to be located in Venezuela to take advantage of low-cost natural gas for hydrogen production and for process fuel.

A single plant integrated with, and adjacent to, a generating station could deliver hot oil, at about 600 F, to the generating station. Since the energy delivered per barrel would be slightly greater in this case, less oil would be required by the power system. This would alter the basic material and heat balances to some extent. In addition, some of the heat exchangers would be eliminated, thus modifying the cost estimates. Using natural gas for fuel instead of desulfurized oil and blending middle distillate back into the product oil would also change the material balance, since less feedstock would be required. Thus, there are six different material balances to be considered. These are designated as:

Plant Number	Credit for Middle Distillate	Natural Gas Fuel	Hot Oil Product
1	Yes	Yes	No
2	Yes	No	No
3	Yes	No	Yes
4	No	Yes	No
5	No	No	No
6	No	No	Yes

The material balances for each of these configurations is shown in Table 13. Numbers 1, 2, 4, and 5 are applicable to a large plant which would supply five 1000-Mw stations, but the material balances for them are expressed in terms of the 1000-Mw station size. Thus, the numbers should be multiplied by 5.0 to convert them to the large-plant basis.

Equipment

For all of the plants, the process equipment would be similar, except in the case where hot oil would be delivered to the generating plant. In that instance, some of the heat exchangers would be eliminated. The only significant difference in equipment would be that in the larger plant all process equipment would be required to have five times as much capacity as in the smaller plant. Regarding auxiliary equipment, tankage requirements would be different for each of the three basic plants, depending on the fuel storage requirement and plant location. These storage requirements are indicated in Table 14 which is the equipment listing for the process. The hydrogen plant and sulfur-recovery plant were assumed to be packaged units, and no equipment breakdown is required.

Capital Cost

Purchased equipment cost estimates were based on generalized cost correlations available in the literature for specific items of equipment and adjusted to an ENR Index of 1270 (mid-1969). Costs for installation, piping, instrumentation, etc., were calculated as a percentage of equipment costs. These percentage factors are reasonably well established for petroleum plants. Construction costs in Venezuela are estimated to be 105% of US costs, as indicated in Ref. 72. The hydrogen plant and sulfur recovery plant costs were based on quotes supplied by vendors for packaged installations.

The capital cost requirements are summarized in Table 15. The scale-up exponent used for the two plant sizes in the US is 0.72. This could have been lower, were it not for a number of heat exchangers in the large plant that would not be required in the small plant. The total fixed capital requirements for the Venezuelan plant would be somewhat less than for the US plant of comparable size, even after allowing for a 5% greater construction cost. This would be due to the elimination of the large storage requirement for long resid, resulting in a saving of about $4,000,000 in tankage. Resid tankage was eliminated in this case because it was assumed that the Venezuelan plant would be located adjacent to a refinery which would provide the tankage. Product oil storage was assumed to be the same for the two large plants.

Annual Owning and Operating Costs

Net operating costs would be a function of the value of by-products (sulfur, middle distillate, and gasoline) and raw materials (long resid and natural gas). Other costs (power, water, labor, catalyst, etc.) are also required to determine net operating costs, but these are either not very significant or are known to a greater certainty.

The operating costs are summarized in Table 16. The most obvious conclusion that can be drawn from this table is that the large plant could produce desulfurized fuel at an incremental cost which would be about 15¢/bbl less than the small plant. Furthermore, the Venezuelan plant would produce desulfurized fuel 10¢/bbl cheaper than the comparable large East Coast plant. In East Coast plants, assuming a credit for middle distillate would mean a saving of about 8.5¢/bbl, while in the Venezuelan plant the saving would be about 7¢/bbl (due mainly to the lower value of middle distillate in Venezuela). In Table 16, the fixed charges include 14% fixed capital, as well as an allowance for plant overhead and interest on working capital.

The difference in costs when a credit is taken or not taken is not simply the credit value assigned to middle distillate. When no credit is taken for middle distillate, it was assumed that the middle distillate would be blended back into the fuel oil. Consequently, the total plant throughput would be less in order to deliver the required energy to the power plant, and there would be a direct saving in long resid required, approximately equal, in volume, to the middle distillate blended in. Secondary effects would be due to the lower direct costs and credits due to the decreased throughput. However, the major effect would be the saving in long resid which would be equivalent to approximately one-half of the value of the middle distillate. Consequently, the cost difference would be only about one-half the credit shown for middle distillate, since one-half the amount would be saved by reduced long resid requirements.

Comparing the small East Coast plant to a large East Coast plant, it is seen that total labor cost per unit of production would be about twice as large in the small plant, reflecting one of the economies of scale. Total material costs would be about the same, as would be expected, although there would be slight differences in individual items. Long resid costs would be higher in the large plant because cold oil, which would be the output product, would have a lower thermal content than the hot oil produced in the small plant. In the large plant, costs for operating supplies and maintenance supplies would be lower than in the small plant, since these items are based on operating labor and unit investment, respectively (again, the economies of scale are illustrated).

Credit values per unit of production would be essentially the same, except for the steam credit. The small plant would deliver hot oil to the power station, and very little excess steam would be produced. Of course, the steam that would be produced in the large plant would require a larger capital investment than would otherwise be required. Thus, the total difference between the two plants can be attributed almost entirely to the economies of scale.

Lower incremental costs would be anticipated for the Venezuelan plant which would allow access to low-cost natural gas for refinery fuel and hydrogen production. Labor costs were estimated to be about the same per unit of production, but large savings would be made in the cost of long resid and natural gas. For long resid the cost would be less because less resid would be required (natural gas would be used as process fuel) and because it has a lower cost than long resid delivered to the East Coast. The total cost of natural gas for use in the process would also be less even though a larger quantity would be used as a fuel. This is because of the very low unit cost for gas. The approximately 10¢/bbl difference in incremental total operating costs between East Coast and Venezuelan locations would be due almost entirely to the use of low-cost natural gas.

Economics of Other Hydrodesulfurization Processes

Alternatives to the vacuum gas oil hydrodesulfurization process were previously discussed, and three alternative schemes were shown in Fig. 10. The economics for these alternative schemes have been previously examined and are given in detail in Ref. 72. A comparison of the economics derived in this study for the VGO process with those of Ref. 72, when adjusted for similar conditions, show very close agreement. Thus, the economics for the alternative process can be taken from Ref. 61 and adjusted to the conditions of this study with an acceptable degree of confidence. These costs for the alternative processes, B, C, and D of Fig. 10, are compared in Table 17 to process A, the VGO process. As shown in the table, these alternative processes would all result in higher fuel costs and were not considered further. Data presented in this table do indicate, however, the costs for providing low-sulfur fuel oil to first-generation steam plants, should the prescribed sulfur goals established for this study be lowered.

Section III - Second-Generation Fuel Desulfurization Processes

This section contains detailed process descriptions, material balances, capital cost estimates, and annual owning and operating cost estimates for those coal and residual oil desulfurization processes identified in Section I as offering the greatest potential for providing low-cost desulfurized fuel to second-generation advanced power systems. Second-generation coal and oil desulfurization processes must utilize gasification, followed by sulfur scrubbing, in order to achieve the low sulfur emission levels desired for second-generation power stations. Those processes employing direct gasification of the raw fuel in high-temperature, slagging gasifiers were found to offer lowest desulfurized fuel cost, although some processes yielding valuable by-products would also be attractive if the by-products could be sold at current market values.

SECOND-GENERATION COAL GASIFICATION

The dominant cost item for first-generation coal gasification processes was noted in the previous section to be the cost of the gasifier vessels. To make a significant reduction in the cost of these vessels, gasification rates must be increased. This could be accomplished by operating at higher temperatures, above the ash-fusion temperature of the coal. As coal is heated to these high temperatures it tends to swell, soften, and sometimes agglomerate. The changes in the properties of coal, plus the slagging of ash, make operation of fixed-bed gasifiers extremely difficult. Consequently, other gasifier designs such as entrained-flow gasifiers must be employed. Processes based on the use of this type of gasifier are described in the following paragraphs, along with a molten salt gasification process.

Partial Oxidation Process Review

The summary of coal gasifiers presented in Table 1 includes several candidate high-temperature slagging gasifiers. For all of these, process air (or oxygen), steam, and pulverized coal would be fed into one end and flow cocurrently through the gasifier. At the high gasification temperatures employed, in excess of 2000 F, hydrogen and carbon monoxide would be the major products. Chemical equilibrium for the formation of methane, higher hydrocarbons, and tars would not be favored. The chemistry of these slagging processes would be essentially the same, but the gasifiers would differ in construction detail and method of solids handling. The Koppers-Totzek gasifier is a horizontal cylindrical vessel. Steam, pulverized coal, and air (or oxygen) are introduced axially at each end of the vessel, and the products exit through a water-cooled central vertical outlet. Most of the coal ash exits with the product gas, but a portion exits as slag through the bottom of the reactor. The Ruhrgas Vortex, B&W-DuPont, and Rummel Single Shaft gasifiers are vertical, cylindrical vessels. Air (or oxygen), pulverized coal, and steam are fed into the bottom of the gasifier, and synthesis or producer gas exits through the top. Reactants are injected either downward with flow reversal, or tangentially into a slag bath, or tangentially into a cyclone chamber in order to achieve a high degree of turbulence at the bottom of B&W-DuPont, Rummel Single Shaft, and Ruhrgas Vortex gasifiers, respectively. Most of the coal ash is being removed as a slag from the bottom of these gasifiers.

The BCR gasifier is a two-stage gasifier. Coal enters the bottom of the upper stage where it flows cocurrently upward in a stream which is a mixture of steam and 3000 F gas from the lower stage. Gas and unconverted coal exit the upper stage at less than 2000 F. The hot char is then separated from the gas and injected into the bottom of the lower stage where it is gasified with air and steam under slagging conditions. By using two stages, the BCR process achieves some internal heat recovery.

The Texaco, USBM, and IGT gasifiers are downflow cocurrent gasifiers. Reactants are injected into the top of a cylindrical vessel and flow cocurrently to a gas exit near the bottom. Ash forms slag on the gasifier walls and flows downward to a water pool at the bottom of the gasifier. In the water pool, the slag solidifies into small particles which are pumped from the vessel. The majority of the coal ash is removed by means of this pool.

Several serious problems would have to be faced in the design and operation of cocurrent, high-pressure, high-temperature coal gasifiers. Among these would be coal feeding, the promotion of turbulence, refractory life, and ash removal. The feeding of pulverized coal into vessels under pressure has been accomplished by lock hoppers and by pumping slurries with water (Ref. 73). Turbulence can be promoted by proper nozzle design for reactant injection. Good mass transfer and temperature control can also be attained by this method. Satisfactory refractory life under the very harsh conditions which would exist in slagging gasifiers is probably the most significant problem in their design and operation (Ref. 74). However, with the proper incentive for development, satisfactory refractory life should be achieved by the 1980's. Ash removal has been achieved by means of water-cooled lock hoppers and water pools.

For all the cocurrent, high-temperature slagging gasifiers surveyed, capital costs and process designs did not appear to be sensitive to the actual design of the gasifier. For this reason one gasifier, the Texaco partial-oxidation gasifier, was chosen as a model for all advanced partial-oxidation processes. A pilot plant for this gasifier is reported to have operated successfully with coals of many ranks (Refs. 75, 76, and 77). Costs developed for the Texaco partial-oxidation gasifiers should be applicable to other high-temperature, slagging gasifiers. For comparison purposes, a Lurgi dry-ash gasifier was also considered for second-generation application.

While the focal point of gasification processes is the partial-oxidation reactor, it is the peripheral equipment (heat exchangers, air compressors, scrubbers, etc.) which would account for most of the cost and where there would be room for process improvement. The oxidation reactor is simply a refractory-lined pressure vessel fitted with spray nozzles and internally baffled. The oxidation reaction should occur at temperatures over 2000 F in order to provide rapid reaction rates and the production of clean fuel gas containing no tars or unsaturated hydrocarbons and negligible amounts of methane and higher hydrocarbons. At higher temperatures, reaction rates increase, but not markedly enough to compensate for the problems associated with very high temperatures. Thus, 2500 F, or thereabouts, represents a reasonable upper limit for the operating temperature, since little can be saved by decreasing the reactor size due to increased reaction rates.

Approximately 20% of the total heat content of a fuel gas from a cocurrent gasifier would be contained as sensible heat in the hot fuel gas. Obtaining satisfactory process efficiency dictates that most of this heat be recovered in the fuel conversion process or utilized in the power system. Analysis of possible alternative methods of recovering waste heat indicated that the most economical method would be to preheat the gasifier reactants.

If feed preheating were not done, additional fuel would have to be oxidized to provide heat for the reactor, and additional air would have to be compressed to provide the air for combustion. Consequently, the required compression power would be increased, leading to lower process efficiency. In addition, a larger proportion of the energy in the fuel would appear as sensible heat in the product gases, compounding the problem of heat recovery. For the present case, a moving pebble bed heater was selected to preheat compressed air with the sensible heat from the hot fuel gas. This is done because it is desired to preheat the air as much as possible, and it is not desired to use metal gas-gas heat exchangers which would require an expensive alloy construction. Shell and tube heat exchangers were used to extract additional heat from the product gases, after their temperature was brought down to a more reasonable level.

Alternative schemes of heat and power recovery have been examined, and in some cases preliminary calculations have been made for alternate schemes. One possible alternative scheme for heat recovery is to use the high-temperature fuel gas to generate high-pressure steam. This high-pressure steam could be expanded in non-condensing turbine to drive the air compressors. After expansion of the steam, its remaining energy could be put back into the process in the form of heat. The heat would be transformed into useful fuel energy at essentially 100% efficiency, since no heat would be rejected to a heat sink. Some preliminary calculations, however, indicate that this concept would offer only a marginal overall improvement in delivered fuel cost, since it would no longer be possible to preheat the air to as high a temperature, and a large additional investment would be required for high-temperature metal heat exchangers.

Another alternative scheme would use the waste heat from the hot fuel gas in a parallel heat recovery system, rather than in a series system. Although this possibility offers some advantages in temperature difference driving forces, it would mean that all of the heat exchangers would be subject to maximum-temperature gas, and alloy exchangers would be required. Other arrangements for heat recovery may also be considered, although there did not appear to be any scheme that offered a clear-cut advantage over the process selected. While a detailed process optimization may indicate some process modifications, it is believed that the resulting fuel processing costs would not be greatly affected.

Another possible alternative would be to use oxygen rather than air. The costs associated with air compression (fixed charges and direct operating) are estimated to be about 12¢/million Btu. If oxygen were valued at $6/ton, then the cost of oxygen would be 20¢/million Btu. Therefore, the use of oxygen is not indicated. Some small savings in fuel cleanup plant fixed charges could be achieved if compressed air were taken off the power plant compressor. However, the savings are marginal, and the process is complicated due to extensive manifolding and controls required.

Sulfur removal from the product gases could be achieved by absorption in liquid scrubbing systems or on solids. Liquid scrubbing, using a hot carbonate absorption system, could reduce the sulfur content of the acid gases below second-generation goals if the gases were cooled below 250 F. The use of hot ferric oxide as a solid absorbent could also achieve second-generation sulfur levels if the gases were cooled below 1000 F. Both these options for sulfur removal were investigated.

Partial Oxidation of Coal for COGAS Power Systems

Several combinations of gasifiers and sulfur removal systems were investigated in order to determine the more economical processes for supplying the fuel requirements of a second-generation, COGAS power system: (1) Texaco partial oxidation with sulfur removal by hot-carbonate scrubbing; (2) Texaco partial oxidation with sulfur removal by hot-ferric oxide absorption; and (3) Lurgi dry-ash gasification with sulfur removal by hot-carbonate scrubbing. The fuel requirement for a second-generation, 54.5% efficient, COGAS power system would be 104.6 million Btu/min (sensible heat plus heating value) of fuel gas at 206 psia (14 atm).

Texaco Partial Oxidation Processes

A flow sheet for a partial oxidation process to produce fuel gas, based on the use of Texaco gasifiers, is shown in Fig. 11. The gasification system would consist of four major sections: coal preparation, coal feeding, gasification, and heat recovery plus a fifth, sulfur-removal section.

Basic Process Description

Coal would be transported from a 90-day storage pile at the rate of 10,228 lb/min, or 7364 tons/day, by means of an underfeed system of two conveyors. These would feed a continuous belt conveyor which in turn would feed four primary crushers in parallel. Minus one inch coal from the crushers would then feed four hammer mills operating in parallel to produce a size consist suitable for gasification (70% through 200 mesh). The crushed coal would be elevated to four storage hoppers from which it would be distributed on a belt conveyor system to four separate process trains, operating in parallel.

The Texaco partial oxidation gasifiers would employ a slurry feeding system. The crushed coal would be mixed with water and steam in agitated slurry tanks to form a 50% slurry with water. The resulting slurry would then be pumped up to the gasifier operating pressure level, 236 psia. The slurry would be vaporized by heat exchange with the hot product gas in shell and tube heat exchangers. The resulting steam-coal ratio would be adjusted from 50:50 to 40:60 in a cyclone-type skimmer with the skimmed steam from the cyclone being recycled to the slurry tank. The steam-coal mixture would be preheated to 1000 F by a second heat exchange with the hot product gas.

Experimental studies by the Bureau of Mines (Ref. 74) indicated that fouling of the slurry vaporizer might be a serious problem in operating a slurry-feed system. Consequently, a lock hopper feeding system was also investigated. Costs for both systems appear comparable.

Process air, approximately 279,000 ft^3/min, would be compressed to 236 psia by two 24,000-hp, 3-stage, intercooled, centrifugal compressors. Combustion air would be preheated in pebble bed heat exchangers (Ref. 78) to the desired preheat temperature of 1900 F. In these heat exchangers, alumina pebbles would form a downward-moving bed. In the upper section of the exchangers, 2200 F fuel gas would be used to preheat the pebbles to 2050 F. The fuel gas would flow countercurrently to the pebbles, and be cooled to 600 F. Hot pebbles would then flow through a seal leg to the lower section of the pebble exchanger where they would preheat countercurrently flowing air to 1900 F. Cooled pebbles at 450 F would be recycled from the bottom of the pebble beds to the top by a bucket elevator.

Preheated steam and coal at 1000 F and air at 1900 F would then flow into the four 11.75-ft-internal-diameter Texaco gasifiers. Each gasifier would be line with 18 in. of refractory material and would be approximately 35.5 ft high. Approximately 95% of the coal would be gasified during the 3 sec residence times, and 5% would be entrained with the slag and lost. Fuel gas would exit the reactor at 2200 F, and be essentially at equilibrium with respect to the water-gas shift reaction. By analogy with cyclone furnaces, it is estimated that approximately 85% of the coal ash would be trapped as slag in the gasifier. The slag would coat the walls of the reactor and flow by gravity to a water-filled pool at the reactor bottom. Slag would be granulated by quenching in the water and would be removed as a finely divided suspension. Hot fuel gas at 2200 F would exit the reactor about two-thirds of the way down and would be used to preheat the coal, steam, and air feeds. After preheating, the fuel gas at 600 F would be available for final dust and sulfur removal.

Process with Hot Carbonate Scrubbing System

A flow sheet for dust and sulfur removal by the hot carbonate scrubbing system is shown in Fig. 4. The 600 F fuel gas would pass through a multiple cyclone system to remove the bulk of residual dust and then would be cooled to 300 F in waste heat boilers. Approximately 185,000 lb/hr of 30-psia, saturated steam would be raised in these boilers for use in the stripping columns of the hot carbonate system. The fuel gas would be further cooled to 225 F and scrubbed of residual dust in four venturi scrubbers and water towers. Each tower would be 12.3 ft in diameter.

Cooled, dedusted gas would then pass through four 6.7-ft-diameter absorption columns where the gas would be countercurrently contacted with a 30% solution of hot potassium carbonate. Any COS present in the fuel gas would be hydrolyzed, and the resulting CO_2 and H_2S would be absorbed by the solution. An estimated 98.5% of the H_2S and 75% of the CO_2 would be removed, yielding a clean fuel gas with a sulfur content of 19 g/million Btu, well below the second-generation sulfur goal.

Spent potassium carbonate solution would flow from the absorption towers to four 6.7-ft-diameter stripping towers where the pressure would be reduced to 30 psia and the carbonate solution stripped with steam. Steam in the stripped acid gas would then be removed in an air-cooled condenser. The resulting acid gas would contain approximately 20% H_2S which would be a suitable feed for a Claus sulfur recovery system.

By-product sulfur would be recovered in a standard Claus converter. One-third of the H_2S in the stripped acid gas would be combusted to SO_2 in air and then the SO_2-H_2S mixture catalytically converted to sulfur. A second-generation Claus converter was assumed to be 95% efficient. The 5% unconverted H_2S would be incinerated and vented to the atmosphere. The amount of unconverted sulfur that would be incinerated would be roughly 60 g/million Btu of clean fuel gas. Thus, the total power station effluent would be 19 g/million Btu from the power system exhaust plus 60 g/million Btu from the Claus incinerator. The total would still be below the second-generation sulfur goal.

Increased sulfur removal, down to a level below 5 g/million Btu, could be achieved with a hot-potassium carbonate scrubbing system. Furthermore, the off-gases from the Claus conversion system could be recycled to the hot-carbonate absorption system. By so doing the total sulfur effluent from the power station could be reduced to about 20 g/million Btu. However, the steam required for stripping the carbonate solution would be increased approximately three-fold, resulting in higher costs. Subsequent cost estimates for second-generation systems are based on the higher sulfur emission rate.

Process with Hot-Ferric Oxide Absorption

The ferric oxide process was investigated as an alternative to sulfur removal by hot-carbonate scrubbing. The major advantage of the ferric oxide system is that sulfur can be removed from the fuel gas at 600 F. Therefore, the product gas would not need to be cooled below the temperature obtained after preheating gasification feeds, 600 F. Since about 400 F of additional sensible heat would be contained in the fuel gas, less coal would have to be processed, resulting in a slightly higher thermal efficiency than if carbonate scrubbing were used to remove H_2S.

The use of ferric oxide at temperatures from 600 to 1000 F as a H_2S absorbent has been studied both in England and in this country. A pilot plant processing 2.5 million ft^3/day of coke oven gas was operated in England for several years. Also, the Southwestern Gas Board (Bristol, England) built a 32-million-ft^3/day plant at Exeter to manufacture town gas from raw coke oven gas. This plant was operated for several years with 96% sulfuric acid as a by-product. Pilot work has also been done by the Bureau of Mines (Ref. 79) aimed at reducing absorbent losses by impregnating ferric oxide on fly ash and pumice.

A flow sheet of a hot-iron oxide process that would remove sulfur for a second-generation COGAS plant is shown in Fig. 5. The fuel gas at 600 F and 211 psia would pass through four 12-ft-diameter by 165-ft-high absorbers in parallel. Each absorber would contain four fluidized beds of ferric oxide in series. Gas and ferric oxide would flow countercurrently to one another through the four beds, and the H_2S and COS would be removed. The product fuel gas would contain approximately 30 g/million Btu of fuel gas.

Spent absorbent would be regenerated in four 7.5-ft-diameter by 140-ft-high fluidized beds in parallel. The exit gas from the regenerator would contain about 7% SO_2 and would be suitable feed for a contact sulfuric acid plant. The regenerated absorbent would be recycled to the absorber. The reactions in this type of absorber are mildly endothermic, but the reactions in the regenerator are strongly exothermic; the regenerator would operate at a temperature about 400 F higher than the absorber.

The fuel gas at 610 F and 206 psia would exit the absorber and proceed to a dust removal section. The primary fuel gas dust removal system would be a rotary stream separator (Ref. 80). These units, described in greater detail in Appendix E, are similar to multicyclones, except that a secondary jet of gas would be used to provide the aerodynamic forces required for a high-efficiency separation. Furthermore, they operate under large variations of inlet dust loading with high efficiency. Experimental data reported in Ref. 80 show 100% removal efficiency for 5-micron particles and 90% for 1-micron particles of 2.65-g/cm^3 density. Typically, better efficiencies were achieved with smaller tube sizes (less than 20-in. diameter). An 8-in.-diameter size, which could handle 300 ft^3/min of gas, was the size assumed for this study.

Based on an assumed inlet dust concentration of about 60 g/ft^3, with 50% less than 10 microns, it was estimated that the outlet from the rotary stream separators would contain about 30 ppm of dust with nothing larger than 5 microns. Final cleanup would be achieved by using high-velocity impact filters, which would be expendable, to bring the solids loading down to about 3 ppm.

Material Balances

A summary of raw material requirements and yields for a Texaco partial-oxidation process with sulfur removal by the hot-carbonate process to supply fuel for a second-generation COGAS plant is shown in Table 18. Approximately 1.822 million tons/year coal and 55 Mw of electricity, mainly to drive the gasification air compressors, would be required. Approximately 585,000 ft^3/min of clean fuel gas at 206 psia (14 atm) and 230 F would be delivered to the power system. The gas would have a higher heating value of 175.8 Btu/ft^3 and a sensible heat above 60 F of 3.2 Btu/ft^3 (a total heat content of 179 Btu/ft^3). Approximately 47,000 long tons/year of by-product sulfur would also be produced. Debiting the electrical requirements for the gasification plant, the net electrical output from the power station would be 945 Mw, and the total efficiency of the station (electricity out/coal in) would be 47.6% as noted in Table 19.

A material balance for the process with sulfur removal by hot-ferric oxide absorption would be slightly different from that for potassium carbonate scrubbing. As noted in Table 19, approximately 1.8 million tons/yr of coal would be required to supply the fuel requirements for a second-generation 1000-Mw COGAS power system. The gas would be supplied to the power system at 206 psia and 610 F. It would have a higher heating value of 168.4 Btu/ft^3 and a sensible heat above 60 F of 10.3 Btu/ft^3 (178.7 Btu/ft^3 total). The overall power station efficiency would be approximately 47.8%.

Capital Costs

Capital costs for the two partial-oxidation processes are summarized in Table 19 per unit of net power output from the power station. Contrary to the first-generation processes, major cost categories would be air compression and heat recovery instead of gasification. The iron oxide process would also require costly acid gas removal and sulfuric acid systems. The process with hot-potassium carbonate scrubbing would cost approximately $26,300,000 or $27·8/kw. The iron oxide process would cost $9.1/kw more.

Annual Owning and Operating Costs

Annual owning and operating costs are also summarized in Table 19. Maintenance and capital-related items were estimated at 5% and 14% of fixed capital, respectively. Other costs for labor, utilities, and coal are tabulated with appropriate credits for by-product sulfur and sulfuric acid. Total fuel costs for these two processes are estimated to be 2.35 and 2.59 mills/kwhr, respectively. The net equivalent fuel costs would be approximately 32.6 and 36.3¢/million Btu, respectively. Based on this comparison, the carbonate scrubbing systems would be the most economical.

Lurgi Dry-Ash Process with Hot-Carbonate Scrubbing System

In addition to high-temperature slagging gasifiers, Lurgi dry-ash gasifiers were also investigated as a means of supplying clean fuel gas to a second-generation COGAS plant. Several modifications to the basic flow sheet for the Lurgi process (Fig. 8) would be required for second-generation processes. For example, the second-generation COGAS power system would be more efficient than the first-generation power system, so that less coal would be required (2.1 million tons/yr vs 2.4 million tons/yr). Also more hydrogen sulfide would have to be removed from the product gas to reduce the sulfur content of the gas to 20 g/million Btu. Finally, a turboexpander would be included in the process to reduce the pressure of the product gases from 315 to 206 psia. The recovered power, together with the reduced airflow rate, would serve to decrease the electrical requirements of the gasification process from 72 Mw to 59 Mw and, thus, increase the net electricity generated in the process to 941 Mw.

The final performance and cost estimates for this process are summarized in Table 19. The net power station efficiency, 42.4%, would be less than that for the Texaco partial-oxidation process. The capital costs would be higher than the Texaco process because of the coal briquette required, the lower gasification rates necessitating more gasification vessels, and the increased investment in compressors because of the higher airflow and compression ratio. The final fuel cost, 3.05 mills/kwhr, would be considerably higher than for the Texaco process so that Lurgi gasifiers would not be selected for second-generation fuel processing systems.

Partial Oxidation of Coal for Base-Load Gas Turbine Power Station

A cost estimate was also prepared for a process with Texaco partial-oxidation gasifiers and hot-carbonate scrubbing to supply clean fuel to a second-generation, base-load gas turbine power system. Such a power plant would have an estimated efficiency of 41.3%. Fuel requirements for this plant would be 137 million Btu/min of clean fuel gas at 500 psia (34 atm). To supply these requirements, the fuel process would consume 2.465 million tons/yr of coal and produce approximately 766,000 ft^3/min of desulfurized fuel gas. The process would consume 90 Mw of power, and thus, the total process would yield a net electrical output of 910 Mw. Overall process efficiency, net electricity out/coal in, would be 34.3%. Details for the process to provide a clean fuel gas for a base-load gas turbine would be the same as those shown in Fig. 11 except for overall flow rates and pressure levels.

Raw material requirements, process yields, capital costs, and operating costs are summarized in Table 19. Total fixed capital costs for the process have been estimated at $34.5/kw net compared to $27.8/kw net for a partial oxidation-hot carbonate process to supply clean fuel for a COGAS plant. This cost increase is due to the lower efficiency of base-load gas turbine which requires 31% additional coal to be processed. Changes in overall capital cost due to higher operating pressure level were found to be negligible. Annual owning and operating costs attributable to the fuel processing section of a clean fuels plant were estimated at 3.26 mills/kwhr for a partial-oxidation, hot-carbonate process that would supply fuel for a base-load turbine plant. This compares to 2.35 mills/kwhr for the same type of process that would provide gas for a COGAS plant. Again, the increase is due to the lower efficiency of the base-load gas turbine process.

Molten Salt Gasification for COGAS Power Systems

The M. W. Kellogg Company Molten Salt Gasification process (Refs. 34 and 38) was also investigated as a possible means of producing desulfurized fuel gas for a second-generation COGAS power system because this process features in situ sulfur removal.

Process Description

The molten salt process was designed by Kellogg to produce synthesis gas which could be upgraded to synthetic natural gas. Heat for the endothermic steam-carbon reaction would be supplied by circulating a stream of molten sodium carbonate between the coal gasification vessel and the melt regeneration vessel. In the latter vessel, a portion of the coal would be burned with air to heat the melt and supply heat for the gasifier. Experimental data indicate that the melt, besides acting as a heat-transfer medium, would act as an H_2S absorbent.

The Kellogg molten salt process has been modified to produce a high-pressure, low-Btu fuel gas for COGAS power systems. A schematic flow sheet of the modified process is shown on Fig. 12. Approximately 11,566 lb/min of coal would be fed from a 90-day storage pile through two stages of hammer mills which would grind the coal to 12 mesh size consist. After grinding, the coal would be lock hoppered into the gasifiers. Gasification would occur at 1830 F and 230 psia in a suspension of coal and molten sodium carbonate. Based upon experimental data presented by Kellogg (Ref. 34), the reaction rate was estimated at 10 lb carbon/hr/ft^3 of melt. Therefore, eleven 16-ft-diameter by 40-ft-high gasification vessels would be required. During the gasification of coal by steam and air, sulfur in the coal would be absorbed by the melt. The purpose of the melt in the modified Kellogg gasification process would be merely as a sulfur acceptor, not as a heat transfer medium. For the purpose of this evaluation, it was assumed that none of the sulfur in the coal would be transferred to the fuel gas. Experimental data indicate that this would be a valid assumption (Ref. 34).

A stream of molten salt would be withdrawn from the gasifiers to maintain a 4% carbon and 8% ash inventory in the reactor. The presence of this amount of carbon and ash appears to catalyze the gasification reaction to some extent. The spent melt, 9500 lb/min, would then be regenerated. The stream would first be quenched and dissolved in water and the insoluble ash, and unreacted carbon would be filtered from the solution. The solution would then be recarbonated with CO_2:

$$Na_2CO_3 + H_2O + CO_2 \rightarrow 2NaHCO_3 \qquad (19)$$

$$Na_2S + CO_2 + H_2O \rightarrow 2NaHCO_3 + H_2S \qquad (20)$$

The relatively insoluble sodium bicarbonate would precipitate and be recovered by filtration. Sodium bicarbonate with a small amount of make-up sodium carbonate would then be recycled to the gasifiers. Calcination of the bicarbonate would occur in situ. The H_2S, which would be evolved during the recarbonation of the spent melt, would serve as a feed to a Claus conversion plant.

The CO_2 requirements for melt regeneration would be met by cooling about 55% of the product gas from the gasifiers, and then scrubbing this stream in a hot-carbonate system to recover the CO_2. During the cooling of this stream of product gas, 447,000 lb/hr of steam and 458,000 ft^3/min of air would be preheated to 1000 F. After CO_2 recovery, the cooled and uncooled gas streams would be recombined to yield a stream of 1100 F product gas at 14 atm pressure.

An alternative system was also investigated in which unpreheated air and steam would be fed to the gasifier, and the recycled sodium bicarbonate would be calcined using the sensible heat of a portion of the product gas in an external calciner. In this system, expensive heat recovery equipment would be eliminated, and the fuel gas could be fed to the power system at a temperature of approximately 1800 F. However, considerably more air would be required for additional combustion of coal to provide the additional energy required because of the lower sensible heat of the feeds to the reactor. Thus, a larger compressor section would be required. In addition, the cost of externally calcining the recycled sodium bicarbonate was estimated to be significantly higher than in the original process. Thus, it was judged that this system would offer no significant economic advantages over the one depicted in Fig. 12.

Material Balance

Raw material requirements and yields for the molten salt gasification process which would supply a clean fuel gas to a second-generation 1000-Mw COGAS power system are summarized in Table 20. Approximately 2.13 million tons/yr of coal would be required. Electrical requirements for the gasification process would be 92 Mw to produce a clean gas at a rate of 893,570 ft^3/min at 14 atm and 1100 F. The gas would have a higher heating value of 110.8 Btu/ft^3 and a sensible heat above 60 F of 21 Btu/ft^3 (total heating value 131.8 Btu/ft^3). The total sulfur oxide emissions from the power station would be less than 20 g/million Btu, all of which would be in the Claus plant effluent. The overall efficiency of the process, net power out/total coal in, was estimated to be 40.5%.

Capital Costs

The capital cost estimates for the molten-salt gasification process designed to produce fuel gas are summarized in Table 21. Capital costs were based on those developed by Kellogg for the molten salt process with appropriate modifications because pipeline quality gas would not be produced. The total fixed capital cost for the process was estimated at $48.1 million or $53.0/kw net. Major components of this cost are gasification, including coal preparation, 25%; air compression, 19%; and melt regeneration, 21%.

Annual Owning and Operating Costs

Annual owning and operating costs attributed to the fuel processing plant are also summarized in Table 21. These costs are based on supplying fuel for second-generation 1000-Mw COGAS power station with a net generating capacity of 908 megawatts. Total fuel cost was estimated to be 3.87 mills/kwhr which is not competitive with that for the partial oxidation-hot carbonate process. Capital costs would be over 90% higher, $53.0/kw compared to $27.8/kw, and operating costs would be approximately 65% higher, 3.87 mills/kwhr compared to 2.35 mills/kwhr.

SECOND-GENERATION RESIDUAL OIL GASIFICATION

Residual oil may also be gasified in a partial-oxidation reactor. As discussed in Section I, several companies have developed processes for license. These processes have been developed for converting liquid hydrocarbons into hydrogen or synthesis gas. Thus, most of the technology has been based on the use of oxygen, although some processes have used air, or oxygen-enriched air, in the few cases when the desired product was a low-Btu reducing gas for use in a blast furnace. Only one significant change would be required in the gasification process to produce a fuel gas, and that is to use compressed air as the source of oxygen rather than pressurized oxygen.

Various types of cylindrical and rectangular reactors with fluidized, recirculated, or fixed-bed systems have been studied as well as one with a water-jacket. In addition, various process modifications have been studied as they relate to preheating the feed streams, premixing of the streams, method of introducing reactants and removing products, steam and oxygen ratios, and heat recovery system designs (e.g., Refs. 81, 82, and 83).

However, it is not the purpose of this study to evaluate subtle differences in the processes or equipment, but to make preliminary estimates of what the processes would be capable of and their associated costs. Thus, processes have been selected which have as a basis conditions and equipment similar to those used in commercial operation for the partial oxidation of heavy oil. These processes have been modified to meet the specific requirements of the present study in regard to air compression and heat recovery.

Texaco Partial Oxidation of Oil for COGAS Power System

Process Description

A process flow sheet for the partial oxidation of residual oil is shown in Fig. 13. Residual oil and steam would be preheated to 750 F with heat from the raw product gases and then mixed and fed to the reactor. Air would be compressed to approximately 15 atmospheres in a steam-driven, 3-stage, intercooled, centrifugal compressor and preheated to 1800 F prior to entering the reactor. The partial-oxidation reaction would take place in a refractory-lined reactor at about 2260 F. The reaction mixture would contain about 5% free carbon to prevent deterioration of the reactor linings by the nickel and vanadium constituents of the slag. The hot gases which would come off the reactor would enter a moving pebble-bed heat exchanger and be cooled to 1460 F while heating air to 1800 F. Some of the free carbon and ash would be deposited on the alumina pebbles and be swept back into the reactor with the hot air. However, it is assumed that all of the carbon would stay with the fuel gas for ultimate removal and recycle to the reactor. In any case, there is no net production of free carbon.

The hot gases would be further cooled by raising steam for the reaction from scrubbing water. The scrubbing water would come from the carbon scrubber, where free carbon and ash would be removed from the product gas. The product gases would be cooled to 1170 F in the steam generator and then cooled to 920 F while preheating the residual oil feed to 750 F. The product gas would be further cooled to 250 F by raising steam at 150 psia and 600 F which would be used to provide most of the power required to drive the centrifugal air compressor.

The product gas at 250 F would now be sent to the carbon scrubbing tower, where free carbon and ash would be removed by a water wash in a packed column. The product fuel gas would leave saturated with water at 177 F. Ash would settle in a pool of liquor in the bottom of the scrubber and be periodically removed. The scrubbing liquor, containing the free carbon, would be mixed with heated residual oil. The free carbon particles would be preferentially wetted by the oil. During the subsequent separation of the oil-water mixture, the carbon particles would be concentrated in the oil phase and recycled to the oil preheater. A portion of the liquor phase would be recycled to the steam generator in sufficient quantity to provide the proper steam-to-oil ratio for the gasification reaction. The remainder would be cooled and recycled to the top of the carbon scrubber.

The product fuel gas off from the top of the carbon scrubber would be cooled to 110 F, and condensed water would be removed. The gas would then enter a Sulfinol absorption system for removal of hydrogen sulfide and carbon dioxide. The desulfurized product fuel would be reheated to 200 F by hot water being recycled to the top of the carbon scrubbing tower. A surge vessel would be provided for the product fuel to level out fluctuations in production rates and demand rates before delivery to the power system.

Essentially all of the sulfur would be removed in the Sulfinol absorption system. Based on operating data from a number of plants, the hydrogen sulfide concentration in the fuel gas could easily be reduced to 0.01% from an inlet concentration of about 0.32%. This corresponds to a sulfur level in the fuel gas of about 24 g/million Btu, a value quite close to the ultimate, third-generation goal (20 g/million Btu).

It was assumed that the hydrogen sulfide was selectively absorbed, and almost complete removal achieved while removing 60% of the CO_2. This would lead to an acid gas composition to the Claus plant having a hydrogen sulfide concentration of about 20%. In the Claus plant, a recovery of 95% elemental sulfur was assumed in a 3-stage system, with 5% of the H_2S incinerated to form SO_2 which would be desulfurized up the stack. The sulfur discharged through the stack would be equivalent to 35.1 g/million Btu, for a total emission for the power station of about 69 g/million Btu.

The sulfur emissions from the Claus plant could be further reduced, at little additional cost, by recycling the tail gases as was previously explained for coal processes. It would also be possible to reduce the sulfur content of the fuel gas to below 0.01% by enlarging the Sulfinol absorption towers. Therefore, while this process, as designed, would have an overall sulfur emission of about 60 g/million Btu, it would have the potential of reducing overall sulfur emissions to about 6.5 g/million Btu.

Material Balance

The material balance for the process shown in Fig. 13 is given in Table 22. This balance is for a nominal 1000-Mw power station and is not sized to correspond to any particular power cycle. This fuel plant is referred to as the base plant and was used to develop preliminary process cost estimates for specific power systems.

A summary of the raw material requirements and yields for a partial-oxidation process plant which would provide fuel for a second-generation COGAS power system is given in Table 23. Residual oil requirements would be 6.41 million bbl/yr. Approximately 624,000 ft^3/min of clean fuel gas at 200 psia and 200 F would be delivered to the power system. The gas would have a higher heating value of 164.9 Btu/ft^3 and a sensible heat above 60 F of 2.7 Btu/ft^3 (a total heat content of 167.6 Btu/ft^3). Approximately 23,050 long tons/yr of sulfur could be produced.

Capital Cost

A capital cost estimate for the process is given in Table 24. Total fixed capital cost was estimated to be $19.7 million or $19.9/kw. The major capital cost items were: air compressors - 32%; heat recovery - 13%; scrubbing and sulfur recovery - 12%; and tankage - 6%. Gasification represented only about 5% of the total system cost. The method of cost estimating for residual oil processes is described in Appendix D.

Annual Owning and Operating Costs

Annual owning and operating costs for the residual oil partial-oxidation process are presented in Table 25. Operating costs are based on a power plant producing 990 Mw net of electrical power with a 70% annual load factor. The estimated equivalent fuel cost is 2.74 mills/kwhr or approximately 41.1¢/million Btu.

Texaco Partial Oxidation of Oil for Base-Load Gas Turbine Power System

Cost estimates have been made for a fuel process plant that would supply fuel for a second-generation base-load gas turbine power system. The process flow sheet for this system would resemble that shown in Fig. 13, but the flow rates would be increased to account for the lower gas turbine efficiency.

Material Balance

A material balance for this process is given in Table 23. Approximately 8.36 million bbl/yr would be required to produce 967-Mw net. The gas flow rate would be 813,000 ft^3/min of clean gas. The composition and heating value would be the same as for the COGAS system.

Capital Cost

An estimate of process capital costs is given in Table 24. The total fixed costs would be $24.2 million or about $25.0/kw. Air compression represents about 38% of these costs.

Annual Owning and Operating Costs

The annual owning and operating costs are given in Table 25. The total estimated equivalent fuel cost would be 3.57 mills/kwhr or approximately 30% more than the fuel cost for the COGAS system.

SECOND-GENERATION PROCESSES WITH BY-PRODUCTS

Significant research has been carried out on the conversion of coal and residual oil to other more desirable liquid fuels for various end uses. Three such processes are described below. Each process was designed to derive maximum valuable by-products from the fuel (coal or oil) and to convert the remainder of the fuel into a clean fuel gas. The GASOLINE and COED processes have been selected to yield a synthetic crude oil by-product and char which would be gasified to produce fuel for power generation. A residual oil coking process has been selected to derive a coker gas oil by-product, which would have a greater value than residual oil. The coke would also then be gasified to supply fuel for power generation. For all of these processes, the economics would be very sensitive to the value assigned to by-product oil.

GASOLINE-Based Process

Solvation processes, in which a portion of the coal would be dissolved in a coal-derived solvent, would yield liquid hydrocarbon by-products whose sale could offset the cost of producing a fuel gas from the remaining char. Therefore, the Project GASOLINE process (Ref. 25), developed by Consolidation Coal Company, was investigated. The GASOLINE process was modified to meet the fuel gas requirements of advanced-cycle power stations. The GASOLINE section of the plant would produce two products: char and synthetic crude oil. The char would be gasified in a high-temperature, entrained-flow gasifier to produce fuel gas.

Process Description

A flow sheet of the process is shown on Fig. 14. Approximately 25,100 tons/day of coal would be transported from a 90-day storage pile, dried, and crushed to a fine consist. This coal would then be slurried with a process-derived solvent and heated to about 650 F. Approximately 60% of the coal would be dissolved to form liquid (extract) and gaseous products.

Undissolved solids would then be separated from the solution in hydrocyclones and carbonized to recover solvent; the resulting char (40% by weight of the coal) would be fed to a Texaco partial-oxidation gasifier system. Details of this gasification system would be similar to those previously presented for straight coal gasification.

Following the solids removal, solvent and extract would be separated by distillation, and the solvent would be recycled to the extraction section of the process. The extract would be washed with a dilute solution of acid for the removal of trace impurities of ash and then hydrotreated at 4000 psig over a suitable catalyst to form a synthetic crude oil.

Hydrogen requirements for the hydrotreater would be met by reforming product gases from the extraction and separation sections, and by the partial oxidation of a portion of the partially converted extract which would be withdrawn from the hydroconversion reactors. Approximately 40% of the hydrogen would be made by reforming product gases.

Material Balances

A summary of raw material requirements and process yields for a GASOLINE-based process to supply a clean fuel gas to a second-generation, 1000-Mw, COGAS power system is shown in Table 26. Process requirements would be 6.4 million tons/yr of coal and 125 Mw of electricity. Approximately 585,000 ft^3/min of clean fuel gas at 230 F and 206 psia would be supplied to the power system. The higher heating value of this fuel gas would be 175.8 Btu/ft^3, and the sensible heat above 60 F would be 3.2 Btu/ft^3 (total heat content 179 Btu/ft^3). The fuel gas would contain 20 g/million Btu sulfur. In addition to the fuel gas, 15.4 million bbl/yr of synthetic crude oil would be produced. Other by-products from the plant would include 136,000 long tons/yr of sulfur, 32.7 million lb/yr of tar acids, and 42,300 tons/yr of ammonia.

Capital Costs

A summary of capital costs for a second-generation, Project GASOLINE-based process is shown on Table 27. Costs for the section of the plant associated with the GASOLINE process were taken from estimates prepared by the Ralph M. Parsons Company (Ref. 84). Costs for the char gasification section of the process were estimated from costs previously developed for coal gasification. Total fixed capital requirements for the process were estimated to be $238 million. Major components of this cost were: solvent-extract separation including char carbonization, 11%, hydrotreating, 35%; hydrogen manufacture, 21%; gasification and fuel gas cleanup, 11%; and general facilities, 8%. Fixed capital costs would be $272/kw net.

Annual Owning and Operating Costs

A summary of annual owning and operating costs for the second-generation Project GASOLINE-based process is shown in Table 28. Costs are based on supplying feed fuel to a second-generation 1000-Mw COGAS power system. After debiting process electrical requirements, the net electrical power from the power station would be 875 megawatts. The efficiency of the process considering electrical output only, i.e., electricity out/coal in, would be 12.9%. The overall efficiency, electricity and oil out/coal in, would be 64.5%. Total annual costs attributable to the fuel processing section of the plant were estimated at 14.66 mills/kwhr. This number includes credit for tar acids at 4¢/lb, sulfur at $25/long ton, and ammonia at $35/ton, but does not include credit for oil. Overall process economics would be quite sensitive to the credit taken for by-product oil as is shown in Fig. 15.

In Fig. 15, annual operating costs attributable to the fuel processing plant are plotted against oil credit for two cases. In the first case, no other by-product credits are taken except for oil, and in the second case credit is taken for sulfur, tar acids, and ammonia. For this latter case, annual operating costs range from about 6 to 3 mills/kwhr as oil credit ranges from $3 to $4/bbl, the most probable range of oil credits. To break even with straight coal gasification, oil would have to be valued at about $4.25/bbl. Thus, it is not likely that a Gasoline-based process could be competitive with straight coal gasification.

COED-Based Processes

Another process which yields by-product oil is the COED multistage pyrolysis process (Refs. 19 and 85). In this process, crushed and dried coal is fed to four fluidized beds operating in series. The first-stage bed operates at a temperature slightly below the agglomerating temperature of the coal. As the volatile matter in the coal is driven off in this stage, the agglomerating temperature increases. Partially devolatilized coal or char then passes to the next stage which operates at a higher temperature. Heat for each stage is supplied by using gases from a hotter stage to fluidize a cooler stage and by recycling char from hotter stages to cooler stages. Ultimately, a portion of the char is combusted in oxygen in the last stage to provide heat for the process. Products from the process include char, tar which is hydrotreated to produce a synthetic crude oil, and a high-Btu fuel gas which can be converted to either hydrogen or methane.

Process Description

The COED process has been modified to meet the requirements of a clean fuel plant. Only two products would be required from the modified process, synthetic crude oil and a clean, low-Btu fuel gas. Because of these requirements, oxygen would not be required in the pyrolysis section, and a high-temperature, entrained-flow gasifier would be added to gasify the product char. A flow sheet for a modified COED process to supply clean fuel gas to a second-generation, 1000-Mw COGAS power system is shown on Fig. 16.

In the modified process, approximately 12,000 tons/day of coal would be transported from the 90-day coal storage pile to primary jaw crushers and hammer mills in parallel. Here a size consist required for fluidized-bed pyrolysis (100% - 14 mesh and 15% - 100 mesh) would be produced. The coal would then be dried in fluidized-bed dryers and lock hoppered into a four-stage fluidized-bed pyrolysis section. Two trains of pyrolysis would be required. The pyrolysis section would operate at a pressure of 16 atm.

In the first stage of pyrolysis, coal would be heated to 600 F, driving off some volatile matter. Heat for the process would be provided by the hot fluidizing gas. Off-gases from the first stage would then be water-quenched to recover oil (2% weight of the coal). A portion of noncondensible gases would be recycled to the dryer and the first stage.

The 600-F char from the first stage would then flow by gravity to the second stage of pyrolysis. There it would be heated to 800 F by hot fluidizing gases from the third stage and by recycle char. Particulate matter in the off-gases from the second stage would be removed by an electrostatic precipitator. The gas would then be quenched in water and the condensibles recovered in various separation towers. Second-stage oil (18% by weight of the coal) would be combined with first-stage oil and pumped to raw-oil storage tanks. A portion of the noncondensible gases would be recycled to provide a fluidizing medium for the third stage of pyrolysis. Approximately 25% of the second-stage product gas would be reformed by hydrogen to upgrade the raw COED oil. The remaining gas would become part of the clean fuel gas product.

Char from the second stage of pyrolysis would be transported by gravity to the third stage and heated to 100 F by hot recycle char. The 1000 F off-gases from the third stage would serve as the fluidizing medium for the second stage.

Third-stage char would then flow by gravity to a fourth stage of pyrolysis where it would be partially gasified at 1600 F in air and steam. Off-gases from the fourth stage would be cooled in a waste heat boiler and form part of the clean fuel product.

Char from the fourth stage of pyrolysis at 1600 F would be gasified with 1000 F steam and 1900 F preheated air in a high-temperature, entrained-flow gasifier such as a Texaco partial-oxidation gasifier. Details of char gasification are essentially the same as those of coal gasification previously discussed. The fuel gas from char gasification would be cooled to 600 F by preheating air and steam feeds to the gasifier. The gas would then be cooled to 300 F in waste heat boilers while raising steam required for the acid gas scrubbing plant. Finally, the gas would be water washed to remove residual dust.

The fuel gas from the char gasification would be combined with pyrolysis off-gases from the second and fourth stages and then would flow to a hot-carbonate, acid-gas scrubbing system. An estimated 98.5% of the H_2S and COS, and 75% of the CO_2, would be removed to yield a clean fuel gas with a sulfur content of 27 g/million Btu. Acid gases would be stripped from the hot-carbonate solution by steam, and the sulfur would be recovered in a Claus converter.

The raw oil, which would have been recovered from the first and second stages of pyrolysis, would be filtered to remove residual solids and hydrotreated. In the hydrotreating section, raw oil and hydrogen would be compressed to 3000 psia and preheated to 700 F in externally fired heaters. The preheated oil would then be hydrotreated over a nickel-molybdenate catalyst. During hydrotreating, heteroatoms (oxygen, nitrogen, and sulfur) would be removed, and the viscosity of the raw oil would be reduced. Hydrotreated oil would then be cooled, condensed, and pumped to oil storage. Noncondensibles, consisting mostly of hydrogen, would be recycled.

Make-up hydrogen for the hydrotreating process would be supplied by reforming a portion of the second-stage product gases. These gases contain over 70% methane and higher hydrocarbons, the balance being primarily hydrogen and CO. Since reforming catalysts are severely poisoned by sulfur, the first stage in gas reforming would be a Sulfinol acid gas removal system to reduce the sulfur content of the gases down to a very low level. Stripped acid gases from the Sulfinol system would also flow to the Claus converter. The gases to be reformed would be passed through a zinc oxide sulfur guard chamber, after which the following reaction would take place over a suitable catalyst:

$$CH_4 + H_2O \rightarrow CO + 3H_2 . \qquad (15)$$

Additional steam would be added to the reformed gases and they would be shifted over a suitable catalyst:

$$CO + H_2O \rightarrow CO_2 + H_2. \qquad (16)$$

The shifted gas would be scrubbed in an MEA absorber to remove CO_2, and the resulting hydrogen stream would be compressed for use in the hydrotreating section.

Material Balance

Raw materials and yields for a COED-based process supplying a clean fuel gas to a second-generation COGAS power system are summarized in Table 29. Coal requirements for the process would be 2.92 million tons/yr. Electrical requirements for the process have been estimated at 72 Mw, primarily to run the gasification and pyrolysis air compressors. Approximately 551,000 ft^3/min of clean fuel gas would be supplied to the power plant at 220 psia and 230 F. The gas would have a higher heating value of 186.7 Btu/ft^3 and a sensible heat above 60 F of 3.2 Btu/ft^3 (total heat content 189.9 Btu/ft^3). The sulfur content of the gas would be 27 g/million Btu, well below the second-generation sulfur goal. In addition to clean fuel gas, 3.09 million bbl/yr of synthetic crude oil, and 78,700 long tons/yr of by-product sulfur would be produced. The overall process efficiency, net electrical power plus synthetic crude oil produced divided by total coal input, has been estimated at 51.7%.

Capital Costs

Capital costs for the COED-based process are summarized in Table 30. Estimates of capital costs for the COED Process (Refs. 19, 85, and 86) were used to estimate the cost of the coal preparation, coal pyrolysis, oil filtration, oil hydrotreating, and hydrogen manufacturing sections of the plant. Previously described estimates of coal gasification by the Texaco partial oxidation process were used to estimate costs of char gasification and fuel gas cleanup. Total fixed capital costs for the process were estimated at $58.3 million or approximately $63/kw net. Major capital cost sections of the plant would be pyrolysis, 23%; oil hydrotreating, 14%; hydrogen manufacture, 11%; and char gasification, 26%. Costs associated with the COED process account for nearly 60% of the total capital cost.

Annual Owning and Operating Costs

Owning and operating costs for the COED-based process are also summarized in Table 30. These costs are based on supplying clean fuel to a second-generation, COGAS power station plant with a net electrical output of 928 Mw. Total operating and owning costs attributable to the fuel processing portion of the plant would be 4.43 mills/kwhr before credit is taken for the by-product crude oil. Economics for the process are highly dependent on the credit taken for the by-product crude. The most probable range of oil credits would be between \$3 and \$4/bbl. Because the oil would be highly aromatic, have zero residual contents, and yield excellent gasoline blending stocks for octane improvement, the \$4/bbl price would be possible (Ref. 85).

The effect of oil credit on annual operating costs is shown in Fig. 17. In this figure, costs are plotted against oil credit for two cases, one in which no credit is taken for by-product sulfur and one in which by-product sulfur is credited at \$25/long ton. For the latter case, the most probable range of annual owning and operating costs would be 2.2 to 2.8 mills/kwhr which would have to be compared with the cost of straight coal gasification estimated at 2.35 mills/kwhr. An oil credit of \$3.75/bbl would be required before the COED process would break even with straight coal gasification.

Coking of Residual Oil

The concept of the coking process is to recover valuable by-products from the residual oil and to turn the least desirable portions of the oil into fuel for a power system. In order to accomplish this, the residual oil would be coked in a fluid coking process with yields of 25% coke, 8% gas, and 67% coker gas oil by-product. The coke would subsequently be gasified in a partial-oxidation gasifier. The partial-oxidation gas would be combined with the gas from the coker to provide a gaseous fuel. Costs for the gasification of coke have been extrapolated from the costs for coal gasification, with the exception that the coke would be already preheated since it would be discharged hot from the coker.

Process Description

A simplified flow sheet for this process is shown in Fig. 18. Residual oil and air would be introduced into the fluid coking system, the air burning a portion of the coke to provide the heat for the process. Coke, gas, and by-product oil would be taken off the fluid coker, with the hot coke gasified with steam and air in a partial-oxidation reactor. The gas from the coker would be combined with the gas from the partial-oxidation gasifier, and the combined gas stream desulfurized by a potassium-carbonate system. The acid gases would be sent to a Claus sulfur recovery unit, and the desulfurized gases sent to the power plant. The product gas would have a HHV of 175.2 Btu/ft^3 and a sulfur content of about 30 g/million Btu.

Economics

Capital requirements for the coking operation, as well as energy, utility, labor requirements, and yields are estimated from values given in Refs. 87, 88, and 89. These capital costs were combined with the previously generated costs for gasification of coal and are summarized in Table 31. The costs for coking and residual oil are very sensitive to the value assumed for the by-product, which would range in value between $3 and $4/bbl, depending upon quality of the oil. Figure 19 shows how the net fuel cost would vary with the value assigned to the by-product oil. The costs are shown in cents/million Btu, making them essentially independent of power cycle. Corresponding to the power cycle assumed (second-generation COGAS at 54.5%), the fuel costs can be converted to power costs in mills/kwhr by multiplying by 0.0897.

Purchase of Petroleum Coke

In reality, a coking process would be a small refinery operation wherein oil would be the principal product and fuel gas would be the by-product. Perhaps a more logical way to operate a power station would be to obtain by-product coke from petroleum refineries to use as a starting material. The feasibility of this alternative would have to be determined by a market study. Some problems that militate against obtaining petroleum coke in this manner would be that by-product coke is not made in quantities large enough to satisfy the demand of a 1000-Mw power system at one refinery location. Petroleum refineries operate their cokers for optimum yield of products other than coke, and the optimum yield would vary with market conditions for liquid products. Thus, refineries might be reluctant to enter into long-term contracts to supply coke as this would deprive them of required flexibility in operating these cokers.

Nevertheless, if it is assumed that petroleum coke is available as a starting material, the economics would be very similar to the economics for coal gasification. Fuel delivered to the power plant would have an incremental cost of 15¢ to 20¢/million Btu, or about 1.3 to 1.8 mills/kwhr.

Section IV - Third-Generation Fuel Desulfurization Processes

This section contains detailed process descriptions, material balances, capital cost estimates, and annual owning and operating cost estimates for those coal and residual oil desulfurization processes which offer the greatest potential for providing low-cost desulfurized fuel to third-generation advanced power systems. The analyses of second-generation processes in the previous section indicated that the most economical processes for producing a clean, desulfurized fuel would involve fuel gasification in high-temperature entrained-flow vessels followed by liquid scrubbing to remove sulfur compounds. Consequently, this type of fuel desulfurization process was also examined for third-generation power stations. In addition, the high-temperature (1400 F to 2000 F) Dolomite process was investigated for third-generation sulfur removal. These processes were designed for use with advanced gas turbine and COGAS power systems.

THIRD-GENERATION COAL GASIFICATION

Third-generation coal gasification processes are based on projections of those processes designed for second-generation systems. Other processes reviewed for second-generation applications, but discarded because of poor performance or high costs, were not considered for third-generation application. Since direct gasification of coal in high-temperature, slagging, entrained-flow gasifiers, such as the Texaco-type partial-oxidation gasifiers, appeared to offer such a substantial economic advantage over other methods of gasification, this was the only type of gasifier considered for third-generation applications.

Partial Oxidation for COGAS Power System

The basic chemistry and mechanics of the partial-oxidation gasification scheme would be the same as those previously described for second-generation plants and are not repeated. The only significant exception would be that third-generation systems operate at a 70% higher pressure. Variations in process design, material balance, and economics due to consideration of calcined dolomite or hot-carbonate sulfur removal schemes are described below.

Calcined Dolomite Sulfur Removal

A flow sheet for a third-generation, partial-oxidation, calcined dolomite process is shown in Fig. 20. Coal, at a rate of 8474 lb/min, would be transported from a 90-day storage pile to four trains of primary jaw crushers and hammer mills in series. Here the size consist of the coal would be reduced to 70% - 2 mesh. The coal would be lock hoppered into a 164,000-lb/hr stream of 350-psia/880-F steam, which would be raised in the power plant section of the process. The resulting steam/coal mixture, along with approximately 223,000 ft^3/min of preheated air, would be fed to two trains of 12-ft-diameter by 36-ft-high gasifiers, in which the coal would be gasified at 2200 F and 345 psia. Hot off-gas would be used to preheat air to 1900 F in two trains of 23-ft-diameter by 55-ft-high pebble-bed heat exchangers. The resulting 1350 F off-gas would then be cooled to a 1150 F by heat exchange with the steam-coal feed.

The fuel gas would flow to two trains of 20-ft-diameter by 20-ft-high fluidized-bed desulfurizers. Here the gases would contact with 28-mesh calcined dolomite, and reactions (14), (19), and (20) would occur. These reactions are exothermic and serve to reheat the gas to a temperature of 1450 F. This is near the lowest temperature at which these reactions would proceed at a reasonable rate. Because the equilibrium for H_2S removal is most favorable at low temperatures, these conditions would correspond to the maximum obtainable H_2S removal. It is estimated that 63% of the CO_2 would be removed by reaction (20), and that 98.5% of of the H_2S would be removed by reaction (18) (Ref. 23). A 30% active calcined dolomite and a fluidizing velocity of 5 ft/sec were the conditions assumed for the desulfurizer.

Spent dolomite from the desulfurizers would flow by gravity to one 7-ft-diameter by 20-ft-high fluidized-bed regenerator. Here the $CaS \cdot MgO$ would be regenerated by reaction (21). A temperature of 1100 F would be maintained in the regenerator by raising steam in the internal coil. The maximum temperature for which the equilibrium for reaction (21) enables a stream of 20% H_2S to be produced would be 1150 F. It was assumed that reaction (21) would proceed to 90% of equilibrium in the regenerator.

Regenerated dolomite would be transported pneumatically to one 14-ft-diameter by 44-ft-high fluidized-bed calciner. At 1900 F, reaction (19) would be reversed to yield active, calcined dolomite for recycle to the desulfurizer. The heat for this reaction would be supplied by combusting approximately 7% of the desulfurizer off-gas in air. Calcined dolomite losses of 2% per cycle were assumed; thus 11 tons/hr of make-up dolomite would be added to the calciner feed.

Off-gas from the regenerator, containing 26% H_2S (dry basis) would pass to a Claus converter system where it would be converted to sulfur with a 90% efficiency. Off-gas from the Claus converter would be combined with approximately 12% of the off-gas from the fluidized-bed calciner and passed through a hot-carbonate scrubbing system which would recover CO_2 for use in the regenerator. It would also remove unconverted H_2S from the Claus effluent.

The remainder of the off-gas from the calciner would be combined with the desulfurizer effluent, raising its temperature to 1500 F, and passed to a dust removal system. High-temperature dust removal could be effected by a number of systems, i.e., multicyclones, electrostatic precipitators, or bag filters.

For the present application, an experimental system was selected similar to that previously described to dedust the effluent from the second-generation partial-oxidation system with hot iron oxide cleanup system (Appendix E).

Material Balance

A summary of raw material requirements and yields for a partial-oxidation, dolomite process which would supply clean fuel gas to a third-generation COGAS power system is shown in Table 32. The COGAS power system would have an efficiency of nearly 58% and operate at 20 atm pressure. Approximately 1.6 million tons/yr of coal would be required to feed the process. In addition, approximately 164,000 lb/hr of 800 F/350 psia steam would be required. This steam would be supplied from the waste-heat boiler of the COGAS power system, resulting in a decrease of approximately 10 Mw from the 1000-Mw nominal power system output. Approximately 67,000 tons/yr of make-up dolomite would be required. The power requirement of the fuel cleanup system would be 59 Mw. Clean fuel gas would be supplied to the power station at 1500 F and 325 psia (22 atm), at a flow rate of about 508,000 ft^3/min. The gas would have a higher heating value of 156.2 Btu/ft^3 and a sensible heat above 60 F of 37.8 Btu/ft^3 (total heat content 194 Btu/ft^3). The fuel gas would have a sulfur content of 16 g/million Btu, a value below the third-generation goal of 20 g/million Btu. In addition to clean fuel gas, approximately 40,000 long tons/yr of by-product sulfur would be produced. The overall efficiency of the process, electricity out/coal in, would be 56.1%.

Capital Costs

Capital costs for a third-generation, partial-oxidation dolomite process are summarized in Table 33. Fixed capital costs were estimated at $27.6 million or $29.9/kw net. Major components of this cost would be sulfur removal, 25%; air compression, 25%; heat recovery, 18%; and gasification, including coal feeding, 12%. A breakdown of the costs for the various sections of the dolomite sulfur removal system is given in Table 34. Costs are presented in dollars per million ft^3 per day of clean gas delivered to the power system. Two trains of desulfurization and one train each of regeneration, calcination, CO_2 recovery, and CO_2 recycle were assumed.

Annual Owning and Operating Costs

Annual owning and operating cost for third-generation partial-oxidation dolomite process are shown in Table 33. The costs are based on supplying clean fuel gas to a COGAS power station with a net electrical output of 922 Mw. With dolomite charges of $5/ton, coal costs of 20¢/million Btu, and a sulfur credit of $25/long ton, the total costs attributable to the fuel processing section of the plant would be 2.41 mills/kwhr. If no credit were taken for by-product sulfur, the total cost would increase by about 7%.

Hot-Carbonate Sulfur Removal

Costs were also developed for a high-temperature, entrained-flow gasification process (assumed to be the Texaco partial-oxidation process) with sulfur removal by the hot-carbonate, liquid scrubbing process. This process would be essentially identical to the one developed for second-generation plants. The only difference between second- and third-generation processes would be the use of a more complicated sulfur removal scheme to reduce the total sulfur effluent from the plant below the third-generation goal of 20 g/million Btu. This scheme, which was described for second-generation sulfur removal processes, would involve recycling the tail gas from the Claus converter, which would contain the sulfur equivalent of 60 g/million Btu of fuel gas, back through the hot-carbonate absorption system. In this manner, the total sulfur effluent from the power station would be reduced to approximately 6.5 g/million Btu.

A flow sheet for the modified sulfur removal system for a partial oxidation process supplying clean fuel gas to a third-generation 1000-Mw, COGAS plant is shown in Fig. 21. The fuel gasification section of this plant would be identical to the one developed for the second-generation fuel cleanup process and shown in Fig. 11. The only changes from Fig. 11 are: (a) less coal (9632 lb/min compared to 10,228 lb/min) would be required because a third-generation power conversion system would be more efficient; and (b) the gasification pressure level would be 24 atm rather than 16 atm.

The fuel gas would be cooled to 600 F by preheating gasifier feeds and flow through a cyclone system for primary dust removal. The gas would then be cooled to 300 F in waste heat boilers, raising 374,000 lb/hr of 30-psia, saturated steam which would be used in acid gas stripping. The cooled gas would be scrubbed in two trains of venturi scrubbers and 13.8-ft-diameter packed towers in series. Here the gas would be cooled to 240 F, and the residual dust would be removed.

The dedusted gas would be combined with recycled tail gas from the Claus converter and desulfurized by scrubbing with a 30% solution of potassium carbonate. Absorption would be accomplished in two trains of 7.5-ft-diameter absorption columns. All the COS, 99.5% of the H_2S, and 55% of the CO_2 in the combined feed gas would be removed by scrubbing. Absorber off-gas at 230 F and 22 atm would flow to the power-generating section of the plant.

Spent potassium carbonate solution would be stripped with approximately 213,000 lb/hr of steam in two 7.5-ft-diameter stripping towers. The stripped gases would be cooled in an air-cooled condenser, and the resulting gas containing 11% H_2S would be processed in a Claus converter.

For the third-generation process, it was assumed that the Claus converter could operate with an 11% H_2S feed and that 90% recovery of the sulfur from the H_2S would be possible. Tail gases from the Claus converter, containing 17% nitrogen, 1% H_2S, and 76% CO_2, would be compressed to 22 atm in a motor-driven, 3-stage, centrifugal compressor and recycled to the absorption tower. These recycled gases would form approximately 6% of the total feed to the hot-carbonate system.

Material Balance

Raw material requirements and yields for a partial-oxidation, hot-carbonate process supplying clean fuel gas to a third-generation 1000-Mw, COGAS power system are summarized in Table 35. Coal requirements for the process would be 1.77 million tons/yr, and 67 Mw of electricity would be required, mainly to drive the gasification air compressors. Approximately 557,000 ft^3/min of clean fuel gas at 230 F and 325 psia (22 atm) would be supplied to the COGAS plant. The gas would have a higher heating value of 173.9 Btu/ft^3 and a sensible heat above 60 F of

3.2 Btu/ft^3 (total heat content 177.1 Btu/ft^3). The sulfur content of the fired gas would be 6.5 g/million Btu. This would also be the total sulfur effluent for the power station, a value roughly one-third the third-generation goal. It would also be one-third the sulfur content in the fuel from a partial-oxidation dolomite process. In addition to the fuel gas, approximately 47,000 long tons/yr of sulfur would be produced. The overall efficiency of the power station, electricity out/coal in, would be 49.9%.

Capital Costs

Capital costs for a third-generation partial-oxidation-hot carbonate process to provide clean fuel gas for 933-Mw COGAS power station are summarized on Table 36. Total fixed capital costs were estimated to be $26.4 million or $28.3/kw net. The major components of the capital costs would be: air compression, 26%; heat recovery, 24%; and acid gas removal, including tail gas recycle, 13%. This estimated cost is approximately 5% lower than the cost developed for the third-generation partial-oxidation dolomite process and is approximately equal to the cost developed for the second-generation partial-oxidation hot-carbonate process. However, the sulfur effluent from the third-generation partial-oxidation hot-carbonate process would be significantly lower than either of the others.

Annual Owning and Operating Costs

Annual owning and operating costs for partial-oxidation hot-carbonate process supplying clean fuel to a third-generation COGAS plant are shown in Table 37. The total annual costs attributable to the fuel processing section of the power station were estimated at 2.33 mills/kwhr. This is roughly 5% lower than the cost for a third-generation partial-oxidation dolomite process, and approximately the same as the costs developed for a second-generation partial-oxidation hot-carbonate process. If no credit for by-product sulfur were taken, fuel costs would be increased by about 9%.

Partial Oxidation for Base-Load Gas Turbine Power System

Raw material requirements and yields, capital costs, and annual operating and owning costs for a partial-oxidation, hot-carbonate process to supply clean fuel for a third-generation base-load gas turbine power system are summarized in Tables 38, 39, and 40, respectively. The base-load gas turbine power system would operate at 43.6% efficiency and at a pressure of 34 atm.

Material Balance

Approximately 2.36 million tons/yr of coal would be required to fuel the gas turbine power system. Electricity requirements for the fuel processing plant would be 100 Mw, primarily to drive the gasification air compressors. Clean fuel gas at a rate of 740,000 ft^3/min, at 230 F and 530 psia (36 atm) would be supplied to the power-generating section of the plant. The gas would have a higher heating value of 173.9 Btu/ft^3 and a sensible heat above 60 F of 3.2 Btu/ft^3 (total heat content 177.1 Btu/ft^3). The sulfur content of the fuel gas would be 6.5 g/million Btu; this would also be the total sulfur effluent from the power station. In addition to clean fuel gas, 62,500 long tons/yr of sulfur would be produced.

Capital Costs

Fixed capital cost for the 900-Mw net power station is estimated to be $31.7 million or $35.2/kw net. This is approximately 25% higher than the cost for a comparable fuel plant to supply a 1000-Mw COGAS power system. The increase is due primarily to the lower efficiency of the base-load gas turbines. These costs are also about 2% higher than those developed for the second-generation base-load gas turbine station. However, the sulfur effluent from the third-generation station would be more than an order of magnitude lower than from the second-generation station.

Annual Owning and Operating Costs

Total annual costs attributable to the fuel processing section of the 900-Mw, third-generation base-load gas turbine power station were estimated at 3.06 mills/kwhr. This is approximately 30% higher than those costs developed for a similar system to supply fuel to COGAS stations. Again, the difference is due primarily to lower efficiency of the base-load turbine plant. The overall efficiency of this station would be 36.2% compared to 49.9% for the COGAS station. Fuel costs for the second- and third-generation base-load gas turbine station are identical, but the third-generation plant has more than an order of magnitude lower sulfur effluent.

THIRD-GENERATION RESIDUAL OIL GASIFICATION

Partial-Oxidation Processes for COGAS and Base-Load Gas Turbine Power Systems

The third-generation partial-oxidation process for gasification of residual oil resembles the second-generation process described in Section III and shown in Fig. 12. Only the flow rates and operating pressures have been modified to meet third-generation power system requirements.

Material Balance

A summary of the raw material requirements and yields for third-generation fuel systems that would supply both nominal 1000-Mw COGAS and base-load gas turbine power systems are given in Table 38.

Capital Cost

Estimates of the capital cost for fuel systems to supply both nominal 1000-Mw COGAS and base-load gas turbine power systems are given in Table 39. The COGAS system would have only 77% of the cost of gas turbine fuel supply systems, i.e., \$19.3/kw vs \$25.1/kw.

Annual Owning and Operating Costs

Estimates of the annual owning and operating costs for fuel processing systems to supply both COGAS and base-load gas turbine power systems are given in Table 40. The annual owning and operating cost for the COGAS system, at 2.59 mills/kwhr, is approximately 74% of the cost for the gas turbine system.

TABLE 1

REPRESENTATIVE COAL GASIFICATION PROCESSES SURVEYED

Process Name	Description	Status	Press. Atm.	Gasifier Outlet Temp., F	Btu of Gas (HHV) / Btu of Coal (HHV)	Sensible Heat of Gas / Btu of Coal (HHV)	Gasification Rate	
							lb/hr-ft^2	lb/hr-ft^3
1. Wellman - Galusha	Fixed bed, counter-current flow	Commercial	1	1250	0.82	0.13	112.4	-
2. Power Gas	Fixed bed, counter-current flow	Commercial	1	1250	0.85	0.13	63	-
3. Bamag - Winkler*	Fluidized bed	Commercial	1	1600	0.66	0.10	170	14
4. Ruhrgas Vortex	Cocurrent, up flow, slagging	Commercial	1	1900	0.66	0.27	-	1.3
5. Lurgi Dry Ash	Fixed bed, counter-current flow	Commercial	20	950	0.82	0.09	300	-
6. Koppers Totzek*	Cocurrent, tangential flow, slagging	Commercial	1	2000	0.67	0.19	-	16
7. B&W - DuPont*	Cocurrent, up flow, slagging	Commercial	1	2200	0.68	0.20	-	24
8. Rummel Single Shaft*	Cocurrent, up flow, slagging	Commercial	1	1800	0.79	0.15	-	21
9. Texaco Partial Oxidation	Cocurrent, down flow, slagging	Pilot	14	2200	0.72	0.19	-	57
10. USBM Pressure Gasifier*	Cocurrent, down flow, slagging	Pilot	20	2500	0.64	0.11	-	430
11. IGT Pressure Gasifier*	Cocurrent, down flow, slagging	Pilot	5	2700	0.83	-	-	40
12. BCR 2-Stage Gasifier*	Cocurrent, up flow, slagging	Pilot	68	1700	0.74	0.07	-	58
13. Kellogg Molten Salt	Salt bath	Pilot	28	1800	0.82	0.08	-	40

---

*Oxygen used rather than air.

TABLE 2

COMPARISON OF HEAT RECOVERY OPTIONS FOR ADVANCED GASIFICATION PROCESSES

	Second Generation		Third Generation	
Capital Cost, $/kw	Preheat Feeds	Raise Steam	Preheat Feeds	Raise Steam
Heat Recovery	8.5	5.0	8.1	4.8
Air Compression	6.6	7.6	6.3	7.2
COGAS (1)	53.0	45.4	51.0	43.1
Steam System(2)	-	14.0	-	14.0
TOTAL	68.1	72.0	65.4	69.1
Annual Cost, mills/kwhr				
Coal @ 20¢/million Btu	1.39	1.47	1.29	1.38
Compressor Power @ 5 mills/kwhr	0.28	0.33	0.26	0.31
Capital Costs @ 14%	1.55	1.65	1.49	1.57
Maintenance @ 5%	0.56	0.59	0.53	0.57
TOTAL	3.78	4.04	3.57	3.83

---

(1) Costs per incremental output for COGAS systems estimated to be $53/kw and $51/kw for second- and third-generation systems, respectively.

(2) Incremental costs for steam systems, exclusive of boiler, estimated to be $65/kw.

TABLE 3

COSTS COMPARISON FOR H-OIL DESULFURIZATION OF RESIDUAL OIL

Feed	Venezuelan			Middle East
Sulfur in Feed - %	2.6			3.9
Sulfur in Product - %	1.0	0.6	0.3	1.0
Cost; Total-¢/million Btu	40.7	43.6	47.0	40.0
Incremental-¢/million Btu	10.7	13.6	17.0	10.0
Incremental-¢/million Btu/% Sulfur	6.7	6.8	7.4	3.5

TABLE 4

SUMMARY OF COMBINATIONS OF FUEL PROCESSING AND ADVANCED-CYCLE POWER SYSTEM

COAL

	1	2*	3*	4	5	6*	7	8*	9*	10	11*	12	13	14
Generation	1	1	1	2	3	2	2	2	3	3	3	2	2	2
Power Plant; Type	COGAS	COGAS	Base Load GT	COGAS	COGAS	COGAS	COGAS	Base Load GT	COGAS	COGAS	Base Load GT	COGAS	COGAS	COGAS
Efficiency	47	47	35.2	54	58	54	54	41.5	58	58	43.6	58	58	58
Fuel Conversion Process	Wellman Galusha	Lurgi	Lurgi	Lurgi	Lurgi	Partial Oxidation	Partial Oxidation	Partial Oxidation	Partial Oxidation	Partial Oxidation	Partial Oxidation	COED	KELLOGG	GASOLINE
Method of Sulfur Removal	Scrubbing	Scrubbing	Scrubbing	Scrubbing	Scrubbing	Scrubbing	Iron Oxide	Scrubbing	Scrubbing	Dolomite	Scrubbing	Scrubbing	Molten Salt	Scrubbing
Fixed Capital - $/kw	82.0	48.0	58.0	43.7	42.0	29.4	39.4	34.5	28.3	29.9	35.2	63.0	53.0	238
Operating Cost - mills/kwhr	4.80	3.98	5.21	3.26	3.07	2.35	2.59	3.06	2.33	2.41	3.06	2.53	3.87	4.60
Combined Fuel and Power System Efficiency	33.0	36.1	26.3	42.4	45.5	47.6	48.4	35.0	49.9	51.6	35.4	----	40.5	----
Sulfur Emission														
g/million Btu	117	220	220	75	6.5	75	30	75	6.5	16	6.5	120	20	20
g/kwhr	0.99	1.74	2.39	0.50	0.04	0.5	0.21	0.66	0.040	0.103	0.055	0.81	0.143	0.143
Potential Sulfur Emission g/kwhr	0.055	0.052	0.071	0.043	0.04	0.043	0.021	0.057	0.04	0.103	0.055	0.044	0.143	0.047

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* Favored process for typical power system and generation.

TABLE 5

SUMMARY OF COMBINATIONS OF FUEL PROCESSING AND ADVANCED-CYCLE POWER SYSTEMS

Residual Oil

	1*	2	3*	4	5*	6	7*
Generation	2	2	2	2	3	3	3
Power Plant; Type	COGAS	COGAS	Base Load GT	COGAS	COGAS	COGAS	Base Load GT
Efficiency	54	54	41.5	54	58	58	43.6
Fuel Conversion Process	Partial Oxidation	Partial Oxidation	Partial Oxidation	Coking & P.O.	Partial Oxidation	Partial Oxidation	Partial Oxidation
Method of Sulfur Removal	Scrubbing	Iron Oxide	Scrubbing	Scrubbing	Scrubbing	Dolomite	Scrubbing
Fixed Capital - $/kw	19.9	29.9	24.8	56.0	19.3	21.0	25.1
Operating Cost - mills/kwhr	2.74	2.88	3.57	0.33	2.59	2.67	3.52
Combined Fuel and Power System Efficiency	51.1	51.5	38.5	----	53.6	55.2	38.9
Sulfur Emission g/million Btu	50	30	50	75	50	16	50
g/kwhr	0.317	0.19	0.422	0.5	0.294	0.095	0.417
Potential Sulfur Emission g/kwhr	0.041	0.19	0.055	0.043	0.038	0.095	0.054

---

* Favored process for typical power system and generation.

TABLE 6

FIRST-GENERATION LURGI GASIFICATION PROCESS RAW MATERIALS AND YIELDS

1000-Mw Nominal Size COGAS System

IN:

Coal - 2.405 million ton/year
13.075 lb/min (as rec.)
144.5 million Btu/min

Electricity - 82 Mw

OUT:

Gas - Temp. - 230 F
Press. - 315 psia
Flow rate - 686,622 ft^3/min
41,943 lb/min
HHV - 172.7 Btu/ft^3
Sensible Heat - 3.3 Btu/ft^3(1)(1)
Sulfur - 221 g/million Btu
Composition, Vol %

H_2O	6.6
H_2	20.9
CO	14.1
CO_2	12.5
CH_4	5.8
H_2S	0.1
COS	0.0
N_2	40.0

Sulfur - 52,067 long tons/year

---

(1) Above 60 F

TABLE 7

CAPITAL AND ANNUAL OWNING AND OPERATING COSTS FOR FIRST-GENERATION LURGI GASIFICATION PROCESS

1000-Mw Nominal COGAS System

	Capital Costs Thousand Dollars
Coal Handling and Preparation	4,700
Gasification, Coal Distribution, Ash Removal, Water Washing, Tar Recycle	20,000
Air Compression	8,900
Heat Recovery and Gas Cooling	2,500
Acid Gas Scrubbing	2,500
Sulfur Recovery	1,900
Subtotal	40,500
General Facilities	4,000
Total Fixed Capital	44,500
Working Capital	1,800
Total Investment	46,300
$/kw	48

	Annual Owning and Operating Costs(1)	
	Thousand Dollars	mills/kwhr
Maintenance @ 5% Capital	2,225	0.40
Labor @ $60,000/man/shift	2,280	0.40
Utilities:		
Cooling Water @ 2¢/thousand gal	294	0.05
Process Water @ 30¢/thousand gal	103	0.02
Steam @ 50¢/1000 lb/hr	358	0.06
Sulfur Credit @ $25/long ton	(1,301)	(0.23)
Subtotal	3,959	0.70
Coal @ 20¢/million Btu	12,217(2)	2.17
Subtotal	16,176	2.87
Capital Related Costs @ 14%	6,230	1.11
Total	22,406	3.98(3)

---

(1) 5.64 x 10^9 kwhr/yr

(2) 0.85 lb/kwhr

(3) 42.1 ¢/million Btu

TABLE 8

FIRST-GENERATION LURGI GASIFICATION
RAW MATERIALS AND YIELDS

1000-Mw Nominal Size Gas Turbine System

IN:

Coal - 3.211 million tons/yr
17,458 lb/min (as rec)
192.9 million Btu/min

Electricity - 109 Mw

OUT:

Gas - Temp. - 230 F
Press. - 315 psia
Flow - 916,778 ft^3/min
56,002 lb/min
HHV - 172.7 Btu/ft^3
Sensible Heat - 3.3 Btu/ft^3(1))
Sulfur - 221 g/million Btu
Sulfur 69,520 long tons/yr

---

(1) Above 60 F

TABLE 9

CAPITAL AND ANNUAL OWNING AND OPERATING COSTS FOR FIRST-GENRATION LURGI GASIFICATION PROCESS

1000-Mw Nominal Gas Turbine System

	Capital Costs Thousand Dollars
Coal Handling and Preparation	5,800
Gasification, Coal Distribution, Ash Removal, Washing, Tar Recycle	24,500
Air Compression	10,900
Heat Recovery and Gas Cooling	3,100
Acid Gas Scrubbing	3,100
Sulfur Recovery	2,300
Subtotal	46,600
General Facilities	4,900
Total Fixed Capital	51,500
Working Capital	2,400
Total Investment	53,900
$/kw	58

	Annual Owning and Operating Costs	
	Thousand Dollars	Mills/kwhr(1)
Maintenance 5% Capital	2,575	0.47
Labor @ $60,000/man/shift	3,044	0.56
Utilities:		
Cooling Water @ 2¢/thousand gal	393	0.07
Process Water @ 30¢/thousand gal	138	0.03
Steam @ 50¢/1000 lb/hr	478	0.09
Sulfur Credit @ $25/long ton	(1,737)	(0.32)
Subtotal	4,891	0.90
Coal @ 20¢/million Btu	16,313(2)	2.99
Subtotal	21,204	3.89
Capital Related @ 14%	7,210	1.32
Total	28,414	5.21(3)

---

(1) 5.46×10^9 kwhr/yr

(2) 1.18 lb/kwhr

(3) 40.4¢/million Btu

TABLE 10

FIRST-GENERATION WELLMAN-GALUSHA PROCESS
RAW MATERIALS AND YIELDS

1000-Mw COGAS System

IN:

Coal - 2.467 million tons/yr
13,409 lb/min (as rec)
148.16 million Btu/min

Electricity - 140 Mw

OUT:

Gas - Temp. - 230 F
Press. - 314 psia
Flow - 742,330 ft^3/min
43,808 lb/min
HHV - 160.4 Btu/ft^3
Sensible Heat - 3.2 Btu/ft^3(1)
Sulfur - 117 g/million Btu
Composition, Vol %

H_2O	6.6
H_2	14.4
CO	27.0
CO_2	2.1
CH_4	2.6
H_2S	0.05
N_2	47.25

Sulfur - 56,830 long tons/yr

---

(1) Above 60 F

TABLE 11

CAPITAL AND ANNUAL OWNING AND OPERATING COSTS FOR FIRST-GENERATION WELLMAN-GALUSHA PROCESS

1000-Mw Nominal COGAS System

	Capital Costs Thousand Dollars
Coal Handling	2,000
Crushing and Briquetting	6,000
Gasification	32,000
Air Blowing	1,300
Off-gas Cooling and Tar Removal	3,800
Off-gas Compression	19,900
Acid Gas Scrubbing	2,400
Sulfur Recovery	2,000
Subtotal	69,400
General Facilities	2,000
Total Fixed Capital	71,400
Working Capital	2,100
Total Investment	73,500
$/kw	82

	Annual Owning and Operating Costs Thousand Dollars	Mills/kwhr(2)
Maintenance @ 5% Capital	3,470	0.66
Labor @ $60,000/man/shift	2,100	0.39
Utilities:		
Cooling Water @ 2¢/thousand gal	511	0.10
Process Water @ 30¢/thousand gal	50	0.01
Sulfur Credit @ $25/long ton	(1,420)	(0.27)
Subtotal	4,711	0.89
Coal @ 20¢/thousand Btu	10,902(2)	2.07
Subtotal	15,613	2.96
Capital Related @ 14%	9,716	1.84
Total	25,329	4.80 (3)

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(1) 5.26×10^9 kwhr/yr

(2) 0.93 lb/kwhr

(3) 46.4¢/million Btu

TABLE 12

INDIRECT DESULFURIZATION OF RESIDUAL OIL

Nominal Material Balance
34,000 BPSD

Stream	1	2	3	5	6	8	9	10	11	13	14	16
lb/hr	525,000	525,000	300,500	223,500	7,410	2,000	62,170	64,170	364,670	302,500	62,170	6,620
BPSD	36,500	36,500	21,900	14,520					21,900	23,300		
P, psi	14.7	14.7	14.7	14.7	350	850	850	850	800	750	750	100
T, F	70	700	500	110	70	110	110	110	680	110	110	110
% S	2.6	2.6	2.04	3.06								
ft^3/min												
H_2						5,100	34,420	39,520	39,520		34,420	118
H_2S						0	5,950	5,950	5,950		5,950	880
CH_4					2,700	100	3,870	3,980	3,980		3,870	94
C_2^+						0	710	710	710		710	127
Total ft^3/min						5,210	44,950	50,160	50,160		44,950	1,289

TABLE 12 CONT'D

Stream	17	18	19	20	21	22	23	24	26	27	28	29	30
lb/hr	295,900	295,900	5,266	104	6,724			4,520	272,350	470,500	25,350	486	18,180
BPSD	22,580	22,580	485						20,600	33,325	1,795		1,480
P	45	45	14.7	45	100				14.7	14.7	14.7		14.7
T	110	620	90	220	100				90	90	90		90
% S	.48	.48	.005						.494	1.65	1.65		.05
ft^3/min													
H_2				0	188	188	0						
H_2S				12	892	1.	891						
CH_4				.5	94.5	94.4	.1						
C_2^+				8.9	135.9	133.0	2.9						
Total				21.9	1310.4	416.4	894.						
Sulfur Long Tons/day								48.5					
SO_2 ft^3/min											45.4		

TABLE 13

MATERIAL BALANCE FOR ALTERNATE PLANTS

1000-Mw Nominal Steam Station

	Credit for Middle Distillate			No Credit for Middle Distillate		
Plant Number	1 Burn Gas	2 Burn Cold Oil	3 Burn Hot Oil	4 Burn Gas	5 Burn Cold Oil	6 Burn Hot Oil
Fuel Prop.						
Oil - lb/gal	8.07	8.07	8.07	8.02	8.02	8.02
Oil API	14.5	14.5	14.5	15.4	15.4	15.4
HHV - Btu/lb	18540	18540	18770	18600	18600	18830
Btu/bbl	6.29×10^6	6.29×10^6	6.36×10^6	6.27×10^6	6.27×10^6	6.35×10^6
C/H Ratio	8.71	8.71	8.71	8.70	8.70	8.70
% Sulfur	1.64	1.65	1.65	1.59	1.60	1.60
Bbl Req. for Power Generation	34,600	34,600	34,200	34,700	34,700	34,300
Btu Required for Fuel/day	8.208×10^9	8.208×10^9	8.948×10^9	8.208×10^9	8.208×10^9	8.948×10^9
Bbl of Oil @ 70%	0	1865	2005	0	1870	2010
Million ft^3/day of gas @ 80%	10.15	0	0	10.15	0	0
Total Bbl of Oil Required	34,600	36,465	36,205	34,700	36,570	36,310
Long Resid Required	36,000	37,940	37,670	34,590	36,470	36,210
Mid Dist.	1460	1539	1528	0	0	0
Gasoline	478	504	500	460	484	481
Long Resid needed for Power	34,500	34,500	34,500	34,500	34,500	34,500
Long Resid Loss	1500	3440	3170	90	1970	1710
Thermal Efficiency $\frac{\text{Long Resid Req'd for Power}}{\text{Long Resid Used}}$	95.8	90.9	91.5	99.7	94.6	95.3

TABLE 14

EQUIPMENT LIST

Plant Size		34,000 BPSD	174,000 BPSD
Process Tanks and Vessels			
V1 - Heavy Gas Oil Surge	CS	3,000 gal	10,000 gal
V2 - Vac. Bottoms	CS	600 bbl	3,000 bbl
V3 - Refined Gas Oil	CS	850 bbl	4,250 bbl
V4 - Middle Distillate	CS	60 bbl	300 bbl
V5 - Sulfinol Inventory	CS	1,000 gal	5,000 gal
V6 - Hydrogen Surge Drum	CS	350 gal	1,750 gal
V7 - High-Pressure Separator	SS Clad	6,000 gal	30,000 gal
V8 - Low-Pressure Separator	CS	6,000 gal	30,000 gal
V9 - Reflux Drum - Fractionater	CS	3,200 gal	16,000 gal
V10 - H_2S Gas Holder	CS	42,000 ft^3	210,000 ft^3
Furnaces			
F1 - Vac Still Heater - Tubes - 5 Cr - 1/2 Mo		237 x 10^6 Btu/hr	1,185 x 10^6 Btu/hr
F2 - Reactor Preheater - Tubes 347 SS		78 x 10^6 Btu/hr	390 x 10^6 Btu/hr
F3 - Fractionator Preheater - Tubes 304 SS		91 x 10^6 Btu/hr	280 x 10^6 Btu/hr
Columns and Towers			
C1 - Vac Still - CS shell and trays		18 ft D x 86 ft H	40 ft D x 86 ft H
C2 - Fractionator - CS shell and trays		6.6 ft D x 60 ft H	14.8 ft D x 60 ft H
C3 - Gas Oil Stripper - CS shell and trays		2.5 ft D x 10 ft H	5.6 ft D x 10 ft H
C4 - H_2S Absorber - CS shell		3.5 ft D x 25 ft H	7.8 ft D x 25 ft H
C5 - H_2S Stripper - CS shell		4 ft D x 20 ft H	9 ft D x 20 ft H
R1 - Reactor 321 SS clad on 1 1/4 Cr - 1/2 Mo		7.5 ft D x 66 ft H	14 ft D x 95 ft H
SE1 - Steam Ejector - 2 stage		5,000 lb/hr steam rate	25,000 lb/hr steam rate
Compressors			
C1 - H_2 Recycle - Centrifugal		650 hp	2 @ 1,625 hp
C2 - H_2 Make Up - Reciprocating		2 @ 500 hp	2 @ 2,500 hp

TABLE 14 CONT'D

Plant Size		34,000 BPSD	174,000 BPSD
Heat Exchangers			
E1 - Vac Reboiler	CS	15,000 ft^2	2 @ 37,000 ft^2
E2 - LVGO Cooler	CS	2 @ 8,000 ft^2	2 @ 40,000 ft^2
E3 - HVGO Cooler	CS	6,700 ft^2	33,000 ft^2
E4 - Reactor Effluent Steam Generator CS Shell, CS Tubes		2,700 ft^2	13,500 ft^2
E6 - Reactor Effluent Exchanger	SS	7,800 ft^2	39,000 ft^2
E7 - Fractionator Cond.	CS	6,600 ft^2	33,000 ft^2
E8 - Reboiler for H_2S Stripper	CS	1,000 ft^2	5,000 ft^2
E9 - Lean Heat Exchangers for Absorber Solution	CS		
(the following heat exchangers are not required for a small plant delivering hot oil)			
E10 - Vac Resid. Steam Generator	CS		1,600 ft^2
E11 - Vac Resid. Cooler	CS		15,000 ft^2
E12 - Feed Exchanger	CS		23,000 ft^2
E13 - Middle Distillate Cooler	CS		9,000 ft^2
E14 - Gas Oil Generator	CS		7,500 ft^2
E15 - Gas Oil Cooler	CS		12,500 ft^2
Tankage			
T1 - Long Resid	CS	765,000 bbl	2,500,000 bbl (360,000 bbl)(Ven)
T2 - Gasoline	CS	15,000 bbl	55,000 bbl
T3 - Middle Distillate	CS	15,000 bbl	55,000 bbl
T4 - Blended Fuel Oil	CS	30,000 bbl	450,000 bbl

TABLE 15

SUMMARY OF CAPITAL COSTS

Thousand Dollars

Plant Size Location	34,000 BPSD East Coast	174,000 BPSD East Coast	174,000 BPSD Venezuela
Total Purchase Equipment	2,803	9,121	
Total Physical Cost	6,441	20,782	
Indirect Costs	2,572	6,856	
Battery Limits; desulfurization	9,013	27,648	
Battery Limits; hydrogen plant	1,602	4,200	
Battery Limits; sulfur recovery	800	2,100	
Total Battery Limits	11,415	33,948	
General Service Facilities @ 15% Bulk Storage, Utilities, and Catalyst at Installed Cost + 15%	1,640	7,913	
Total Fixed Capital	14,767	46,951	45,125*
Working Capital	1,150	6,200	1,720
Startup	571	1,410	1,354
Total Investment (exclusive of Land)	16,508	54,561	48,179

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* 105% of US plant adjusted for bulk tankage.

TABLE 16

INCREMENTAL OWNING AND OPERATING COSTS

Plant Size BPSD	34,000	174,000	174,000
Plant Location	East Coast	East Coast	Venezuela
Total Labor	5.68(1)	2.59	2.53
Long Resid	18.52	19.90	7.59
Natural Gas	6.66	6.59	2.48
Other Materials	3.29	2.26	2.20
Utilities	1.54	1.68	1.68
Total Direct Operating Cost	35.69	33.03	16.48
Fixed Charges	22.10	13.42	12.50
Total Operating Cost	57.79	46.44	28.98
Credits			
Sulfur	(3.49)	(3.47)	(3.47)
Steam	(0.69)	(3.73)	(2.03)
Gasoline	(6.75)	(6.73)	(3.77)
Middle Distillate	(16.90)	(16.82)	(14.15)
Total Credits	(27.83)	(30.75)	(23.42)
Net Operating Cost	29.96	15.69	5.54
Net Operating Cost ¢/million Btu	4.70	2.49	.87
Additional if no credit is allowed for middle distillate	8.44	8.51	7.16
Net Operating Cost	38.40	24.20	12.70
Net Operating Cost ¢/million Btu	6.04	3.84	2.02

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(1) Costs in ¢/bbl except where indicated otherwise, and represent incremental costs, excluding transportation.

TABLE 17

COSTS FOR HYDRODESULFURIZATION

1000-Mw Nominal Size

Process	A	B	C	D
	VGO	VGO & SDA	VGO SDA & Coking	Total Residual Hydrodesulfurization
Total Operating Costs(1), ¢/million Btu	38.4	47.1	49.9	48.4
Mills/kwhr	4.30	5.29	5.59	5.43
Sulfur Content %	1.65	0.73	0.42	0.82
Sulfur Emission g/million Btu	450	200	115	250

---

(1) 6.13×10^9 kwhr/yr

TABLE 18

MATERIAL BALANCE FOR PARTIAL OXIDATION OF COAL

Texaco Gasification with Carbonate Scrubbing
1000-Mw Nominal COGAS System

IN:

Coal - 1.882 million tons/yr
10,228 lb/min (as rec.)
113 million Btu/min

Electricity - 55 Mw

OUT:

Gas: Press. - 206 psia
Temp. - 230 F
Flow: 584,744 ft^3/min
31,849 lb/min
HHV: 175.8 Btu/ft^3
Sensible Heat: 3.2 Btu/ft^3
Sulfur: 19 g/million Btu

Composition	Volume %
H_2O	8.5
H_2	25.3
CO	27.5
CO_2	1.0
CH_4	0.5
H_2S	0.009
COS	0.0
N_2	37.2

Sulfur: 46,811 Long tons/yr

TABLE 19

COMPARISON OF SECOND-GENERATION FUEL CLEANUP PROCESSES

Gasification Process	Partial Oxidation	Partial Oxidation	Lurgi Dry Ash	Partial Oxidation
H_2S Removal	Hot-Carbonate	Iron Oxide	Hot-Carbonate	Hot-Carbonate
Power System	COGAS	COGAS	COGAS	Base-Load Turbine
Coal Requirement, million ton/yr	1.882	1.802	2.103	2.465
Electrical Requirement, Mw	55	89	90	59
Gas Output, ft^3	584,744	585,181	563,834	766,015
Heating Value, Btu/ft^3	179.0	178.7	185.5	179
Pressure, psia	206	206	206	500
Sulfur in fuel gas, g/million Btu	19	30	20	19
Net Electrical Output, Mw	945	911	941	910
Overall Efficiency, %	47.6	47.8	42.4	34.3
Capital Cost, $/kw net				
Coal Preparation	2.2	2.2	4.6	3.1
Gasification	3.9	3.9	14.7	2.7
Air Compression	6.9	6.8	8.5	10.9
Heat Recovery	7.0	6.3	2.4	8.9
Dust Removal	1.7	1.1	---	1.6
Acid Gas Removal	2.6	8.4	3.2	2.7
Sulfur or H_2SO_4 Recovery	1.9	6.3	2.3	2.4
General Facilities	1.6	1.9	5.3	2.2
Total	27.8	36.9	41.0	34.5
Annual Owning and Operating Costs, mills/kwhr				
Maintenance @ 5% Capital	0.23	0.30	0.33	0.29
Labor @ $60,000/man/shift	0.21	0.26	0.34	0.28
Utilities:				
Cooling Water @ 2¢/thousand gal	0.01	0.02	0.04	0.02
Process Water @ 30¢/thousand gal	0.02	0.04	0.03	0.03
Steam @ 50¢/thousand lb/hr	--	--	0.16	--
Sulfur ($25/long ton) or H_2SO_4 ($10/ton) Credit	(0.20)	(0.28)	(0.20)	(0.28)
Coal @ 20¢/million Btu	1.44	1.41	1.61	1.99
Capital Related @ 14% Capital	0.64	0.84	0.95	0.79
Total	2.35	2.59	3.26	3.12
Equivalent Fuel Cost, ¢/million Btu	32.6	36.3	40.5	31.4

TABLE 20

MATERIAL BALANCE FOR KELLOGG MOLTEN SALT GASIFICATION PROCESS

1000-Mw Nominal Size

IN:

Coal: 2,128 million ton/yr
11,566 lb/min (as rec.)

Electricity: 92 Mw

OUT:

Gas: Temp. - 1100 F
Press. - 206 psia
Flow rate: 893,570 ft^3/min
49,453 lb/min
HHV - 110.8 Btu/ft^3
Sensible Heat - 21 Btu/ft^3
Sulfur - 20 g/million Btu
Composition, Vol. %

H_2O	13.8
H_2	15.5
CO	17.2
CO_2	7.2
CH_4	0.5
H_2S	--
N_2	45.8

Sulfur - 59,700 long tons/yr

TABLE 21

CAPITAL AND ANNUAL OWNING AND OPERATING COSTS FOR KELLOGG MOLTEN SALT GASIFICATION PROCESS

1000-Mw Nominal Size

	Capital Cost Thousand Dollars
Gasification	13,100
Heat Exchange	9,200
Air Compression	10,300
Melt Regeneration	11,000
Sulfur Recovery	2,000
Subtotal	45,600
General Facilities	2,500
Total Fixed Capital	48,100
Fixed Capital - $/kw	53

	Annual Owning and Operating Costs	
	Thousand Dollars	mills/kwhr(1)
Maintenance, 5% Fixed Capital	2,405	0.43
Labor @ $60,000/man/shift	1,320	0.24
Makeup Na_2CO_3 @ 1.65¢/lb	2,428	0.44
Utilities		
Cooling Water @ 2¢/thousand gal	208	0.04
Process Water @ 30¢/thousand gal	200	0.03
Steam @ 50¢/thousand lb /hr	307	0.06
Sulfur Credit @ $25/long ton	(1,493)	(0.27)
Subtotal	5,375	0.97
Coal @ 20¢/million Btu	9,403(2)	1.69
Subtotal	14,778	2.66
Capital Related @ 14% Fixed Cap.	6,734	1.21
Total	21,512	3.87(3)

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(1) 5.366 x 10^9 kwhr/yr
(2) 0.793 lb/kwhr
(3) 44.6¢/million Btu

TABLE 22

MATERIAL BALANCE - PARTIAL OXIDATION OF RESIDUAL OIL

Stream	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Oil lb/hr	471,000	471,000															
Ash or Carbon lb/hr			20,500	20,500	20,500	20,500		420	20,500			20,500			20,500		
Water/Steam lb/hr			235,000				510,000							171,000	235,000		218,000
Gas ft^3/min				820,000	820,000	820,000			820,000			820,000	797,000				
Air ft^3/min										458,000	458,000					458,000	
Total lb/hr	471,000	471,000	256,000	2,846,000	2,846,000	2,846,000	510,000		2,846,000	2,120,000	2,120,000	2,846,000	2,761,000	171,000	256,000	2,120,000	218,000
P-psia	14.7	225	225	210	205	200	150	14.7	220	230	230	215	200	200	200	14.7	150
Temp., F	80	750	750	1170	920	250	600	250	2260	1800	290	1460	177	80	250	80	600

Stream	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
Oil lb/hr																41,000	41,000
Ash or Carbon lb/hr																	20,500
Water/Steam lb/hr	771,000	728,000		68,100										600,000	600,000		
Gas ft^3/min			797,000		7/2,000			12,020				760,000	760,000				
Air ft^3/min																	
Total lb/hr	771,000	728,000	2,761,000	68,100	2,693,000	1,578,000	1,578,000	78,700	1,500,000	1,500,000	1,500,000	2,615,000	2,514,000	600,000	600,000	41,000	43,500
P-psia	200	150	200	200	200	200	20	20	200	200	200	200	200	200	200	200	200
Temp., F	120	600	110	110	110	110	125	110	150	135	110	110	200	275	130	275	275

TABLE 23

MATERIAL BALANCE FOR PARTIAL OXIDATION OF RESIDUAL OIL

1000-Mw Nominal Size

COGAS Plant		Base-Load Turbine Plant	
In:	Out:	In:	Out:
Oil:	Gas:	Oil:	Gas:
6.41 million bbl/yr	Temp. - 200 F	8.36 million bbl/yr	Temp. - 200 F
109.9 million Btu/min	Press. - 200 psia	142.9 million Btu/min	Press. - 485 psia
	Flow:		Flow
Electricity	624,000 ft^3/min	Electricity:	813,000 ft^3/min
13 Mw	33,400 lb/min	33 Mw	43,500 lb/min
(equivalent steam)	HHV, 164.9 Btu/ft^3		HHV - 164.9
	Sensible Heat - 2.7 Btu/ft^3.		Sensible Heat - 2.7 Btu/ft^3
	Sulfur - 49 g/million Btu		Sulfur - 49 g/million Btu
	Composition		
	CO 25.8%		
	H_2 24.3%		
	CH_4 0.4%		
	H_2O 0.6%		
	CO_2 0.9%		
	N_2 48.0%		
	H_2S 0.01%		
	Sulfur 23,050 long tons/yr		Sulfur 30,000 long tons/yr

TABLE 24

CAPITAL COSTS FOR PARTIAL OXIDATION OF RESIDUAL OIL

1000-Mw Nominal Size Power System

	COGAS Plant Thousands of Dollars	Base-Load Gas Turbine Plant Thousands of Dollars
Gasification	900	500
Air Compression	6,300	9,600
Heat Recovery	2,600	3,200
Acid Gas Removal	1,500	1,400
Sulfur Recovery	900	1,000
Tankage	1,200	1,400
Off-sites	2,700	2,800
General Facilities	2,000	2,400
Indirects	1,600	1,900
Total Fixed Capital	19,700	24,200
$/kw	19.9	25.0

TABLE 25

ANNUAL OWNING AND OPERATING COSTS FOR PARTIAL-OXIDATION OF OIL

1000-Mw Nominal COGAS and Base-Load Turbine Systems

	COGAS		Base-Load Turbine	
	Thousand Dollars	Mills/kwhr(1)	Thousand Dollars	Mills/kwhr(2)
Maintenance @ 5% Fixed Capital	985	0.16	1,185	0.20
Labor @ 60,000/man/shift	1,221	0.21	1,505	0.25
Utilities				
Cooling Water @ 2¢/thousand gal	13	0.002	17	0.003
Process Water @ 30¢/thousand gal	22	0.004	29	0.005
Raw Materials				
Absorber Make-up	114	0.022	188	0.04
Sulfur Credit @ $25/long ton	(576)	(0.096)	(754)	(0.135)
Subtotal	1,809	0.306	2,052	0.363
Oil @ 30¢/million Btu	12,130	2.00	15,830	2.675
Subtotal	13,939	2.306	17,882	3.038
Capital Charges @ 14%	2,760	0.454	3,390	0.572
Total	16,699	2.76(3)	21,272	3.61(4)

(1) 6.01×10^9 kwhr/yr
(2) 5.8×10^9 kwhr/yr
(3) 41.1¢/million Btu
(4) 41.2¢/million Btu

TABLE 26

MATERIAL BALANCE FOR PROJECT GASOLINE-BASED PROCESS

1000-Mw Nominal Size

IN:

Coal: 6.413 million tons/yr
34,861 lb/min
385.2 million Btu/min

Electricity: 125 Mw

OUT:

Gas: Temp. - 230 F
Press. - 206 psia
Flow: 585,000 ft^3/min
31,900 lb/min
HHV - 175.8 Btu/ft^3
Sensible Heat - 3.2 Btu/ft^3
Sulfur - 20 g/million Btu
Composition, Vol. %

H_2O	8.5
H_2	25.3
CO	27.5
CO_2	1.0
CH_4	0.5
H_2S	0.01
N_2	37.2

Crude Oil: 60,400 bbl/day

Sulfur: 136,000 long tons/yr

Tar Acids: 32.7 million lb/yr

Ammonia: 42,300 tons/yr

TABLE 27

CAPITAL COST FOR PROJECT GASOLINE-BASED PROCESS

	Thousands Dollars
Coal Preparation	9,000
Extraction	15,000
Separation	28,000
Hydrotreating	187,000
H_2 Manufacture	52,000
Gasification	18,000
Gas Clean-up	4,000
Sulfur Recovery	5,000
Total Installed Cost	218,000
General Facilities	20,000
Total Fixed Capital	238,000
$/kw	272

TABLE 28

ANNUAL OWNING AND OPERATING COSTS FOR SECOND GENERATION PROJECT "GASOLINE" BASED PROCESS

	Thousand Dollars	Mills/kwhr(1)
Maintenance @ 5% Fixed Capital	11,900	2.22
Labor @ $60,000/man/shift	3,660	0.68
Raw Materials		
Oxygen @ $6/ton	3,388	0.63
Catalysts	3,484	0.65
Misc. Chemicals	470	0.09
Utilities		
Cooling Water @ 2¢/thousand gal	175	0.03
Process Water @ 30¢/thousand gal	120	0.02
Credits		
Tar Acids @ 4¢/lb	(1,308)	(0.24)
Sulfur @ $25/long ton	(3,400)	(0.63)
Ammonia @ $35/ton	(1,481)	(0.28)
Net Incremental Cost	17,008	3.17
Coal @ 20¢/million Btu	28,343(2)	5.28
Subtotal Cost	45,351	8.45
Capital Related Cost @ 14% Fixed Capital	33,320	6.21
Subtotal	78,671	14.66
Oil Credit @ $3.50/bbl	(54,005)	(10.06)
Total	24,666	4.60(3)

---

(1) 5.366×10^9 kwhr/yr, 875 Mw)

(2) 3.29 lb/kwhr

(3) 17.4 ¢/million Btu

TABLE 29

MATERIAL BALANCE FOR COED-BASED PROCESS

1000-Mw Nominal COGAS System

IN:

Coal: 2.92 million tons/yr
15,900 lb/min (as rec)
176 million Btu/min

Electricity: 72 Mw

OUT:

Gas: Temp. - 230 F
Press. - 220 psia
Flow Rate - 551,000 ft^3/min
33,500 lb/min
HHV - 186.7 Btu/ft^3
Sensible Heat - 3.2 Btu/ft^3
Sulfur - 27 million Btu
Composition, Vol. %

H_2O	7.6
H_2	15.5
CO	28.4
CO_2	1.2
CH_4	2.7
C_2+	0.7
H_2S	0.013
N_2	43.9

Crude Oil - 12,100 bbl/day

Sulfur - 78,700 long tons/yr

TABLE 30

CAPITAL AND ANNUAL OWNING AND OPERATING COSTS FOR COED-BASED PROCESSES

	Capital Cost Thousand Dollars
Coal Preparation	2,500
COED Pyrolysis	13,300
Oil Filtration	1,100
Oil Hydrotreating	8,400
Hydrogen Manufacturing	6,600
Char Gasification	15,400
Fuel Gas Clean-Up	4,300
Sulfur Recovery	2,800
Total Cost	54,400
General Facilities	3,900
Total Fixed Cost	58,300
$/kw	63

	Annual Owning and Operating Costs	
	Thousand Dollars	Mills/kwhr(1)
Maintenance @ 5% Capital	2,915	0.51
Labor @ $60,000/man/shift	2,100	0.37
Utilities		
Cooling Water @ 2¢/thousand gal	190	0.03
Process Water @ 30¢/thousand gal	440	0.08
Steam @ 50¢/ thousand lb/hr	500	0.09
Sulfur Credit @ $25/long ton	(1,968)	(0.35)
Net Incremental Operating Cost	4,177	0.73
Coal @ 20¢/million Btu	12,927(2)	2.27
Subtotal	17,104	3.00
Capital Related Costs @ 14% Capital	8,162	1.43
Subtotal	25,266	4.43
Oil Credit @ $3.50/bbl	(10,820)	(1.90)
Total	14,446	2.53(3)

---

(1) 5.68×10^9 kwhr/yr

(2) 1.029 lb/kwhr

(3) 23.2¢/million Btu

TABLE 31

CAPITAL AND ANNUAL OWNING AND OPERATING COSTS FOR COKING AND COKE GASIFICATION PROCESS

1000-Mw Nominal COGAS System

	Capital Costs Thousand Dollars
Fluid Coke Plant	22,200
Partial Oxidation, including H_2S scrubbing and sulfur recording	27,100
Total Installed Cost	49,300
Tankage	11,400
General Facilities	4,900
Total Fixed Capital	65,600

	Annual Owning and Operating Costs	
	Thousand Dollars	Mills/kwhr (1)
Net Operating Costs (by-product oil not included)	4,130	.675
Indirect Charges	1,160	.189
Fixed Charges	7,410	1.21
Residual Oil	50,900	8.30
Subtotal	63,600	10.87
Credit for By-product Oil (@ $3/bbl)	(61,600)	(10.04)
TOTAL	2,000	.326

---

(1) 6.13×10^9 kwhr/yr

TABLE 32

MATERIAL BALANCE FOR PARTIAL-OXIDATION, DOLOMITE PROCESS

1000-Mw Nominal COGAS System

IN:

Coal: 1.559 million tons/yr
8474 lb/min (as rec)
93.63 million Btu/min

Steam: 1008 million lb/yr
(350 psia, 800 F)
164,400 lb/hr (19 Mw)

Dolomite: 67,230 tons/yr
367 lb/min

Electricity: 59 Mw

OUT:

Gas: Temp. - 1500 F
Press. - 325 psia
Flow Rate - 508,204 ft^3/min
29,955 lb/min
HHV - 156.2 Btu/ft^3
Sensible - 37.8 Btu/ft^3
Sulfur - 16 g/million Btu
Composition, Vol. %

H_2O	7.0
H_2	21.7
CO	25.3
CO_2	4.9
CH_4	0.4
H_2S	0.008
COS	0.023
N_2	40.7

Sulfur: 39,912 long tons/yr

TABLE 33

CAPITAL AND ANNUAL OWNING AND OPERATING COSTS FOR PARTIAL OXIDATION - DOLOMITE PROCESS

1000-Mw Nominal Size

	Capital Costs Thousand Dollars
Coal Preparation	1,800
Gasification	3,300
Compression	6,800
Heat Recovery	5,100
Sulfur Removal	6,900
Sulfur Recovery	1,300
Dust Removal	1,400
	26,600
General Facilities	1,000
Total Fixed Capital	27,600
$/kw	29.9

	Annual Owning and Operating Costs Thousand Dollars	Annual Owning and Operating Costs mills/kwhr (1)
Maintenance @ 5% Fixed Capital	1,380	0.24
Labor @ $60,000/man/shift	1,620	0.29
Utilities		
Cooling Water @ 2¢/thousand gal	180	0.03
Process Water @ 30¢/thousand gal	5	0.00
Raw Materials		
Dolomite @ $5/ton	336	0.06
Filter Media	420	0.07
Sulfur Credit @ $25/long ton	(999)	(0.18)
Subtotal	2,942	0.51
Coal @ 20¢/million Btu	6,890 (2)	1.22
Subtotal	9,832	1.73
Capital Charges @ 14% Fixed Capital	3,864	0.68
Total	13,696	2.41 (3)

---

(1) 5.76×10^9 kwhr/yr

(2) 0.551 lb/kwhr

(3) 35.2 ¢/million Btu

TABLE 34

CAPITAL COST FOR CALCINED DOLOMITE SULFUR REMOVAL

	$\$$/million ft^3/day(1)	% of Total
Desulfurizer	2,700	28.7
Regenerator	200	2.2
Calciner	1,100	11.7
CO_2 Recovery	2,100	22.3
CO_2 Recycle	3,300	35.1
Total	9,400	100.0

---

(1) 730 million ft^3/day of gas at 22 atm and 1500 F

TABLE 35

MATERIAL BALANCE FOR PARTIAL OXIDATION-HOT CARBONATE PROCESS

1000-Mw Nominal Size

IN:
Coal -
  1.772 million tons/yr
  9632 lb/min
  106.4 million Btu/min

Electricity - 67 Mw

OUT:
Gas -
  Temp. - 230 F
  Press. - 325 psia
  Flow: 556,643 ft^3/min
  31,497 lb/min

HHV - 173.9 Btu/ft^3
Sensible Heat - 3.2 Btu/ft^3
Sulfur - 6.5 g/million/Btu

Composition,	Vol. %
H_2O	5.5
H_2	25.0
CO	27.2
CO_2	3.8
CH_4	0.5
H_2S	0.003
N_2	38.0

Sulfur: 46,976 long tons/day

IN:
Coal -
  2.357 million tons/yr
  12,812 lb/min
  144.6 million Btu/min

Electricity: 100 Mw

OUT:
Gas -
  Temp. - 230 F
  Press. - 530 psia
  Flow: 740,447 ft^3/min
  41,897 lb/min

HHV - 173.9 Btu/ft^3
Sensible Heat - 3.2 Btu/ft^3
Sulfur - 6.6 g/million/Btu

Composition,	Vol. %
H_2O	5.5
H_2	25.0
CO	27.2
CO_2	3.8
CH_4	0.5
H_2S	0.003
N_2	38.0

Sulfur: 62,487 long tons/day

TABLE 36

CAPITAL COSTS FOR PARTIAL OXIDATION-HOT CARBONATE PROCESS

1000-Mw Nominal Size

	COGAS Plant Thousand Dollars	Base Load Turbine Plant Thousand Dollars
Coal Handling	2,000	2,400
Coal Feeding	1,200	1,500
Gasification	1,800	1,600
Air Compression	6,900	9,700
Heat Recovery	6,300	7,900
Dust Removal	1,200	1,000
Acid Gas Removal	3,300	3,500
Sulfur Recovery	1,800	2,200
	24,500	29,800
General Facilities	1,900	1,900
Total Fixed Capital	26,400	31,700
$/kw	28.3	35.2

TABLE 37

ANNUAL OWNING AND OPERATING COSTS FOR PARTIAL OXIDATION-HOT CARBONATE PROCESS

1000-Mw Nominal Size

	COGAS Plant		Base Load Turbine Plant	
	Thousand Dollar	Mills/kwhr(1)	Thousand Dollar	Mills/kwhr (2)
Maintenance @ 5% Fixed Capital	1,320	0.23	1.585	0.29
Labor @ $60,000/man/shift	1,200	0.21	1,620	0.29
Utilities:				
Cooling Water @ 2¢/thousand gal	159	0.03	208	0.04
Process Water @ 30¢/thousand gal	196	0.03	196	0.03
Steam @ 50¢/thousand lb/hr	123	0.02	-	-
Sulfur Credit @ $25/long ton	(1,174)	(0.21)	(1,565)	(0.28)
Net Increment Cost	1,824	0.31	2,047	0.37
Coal @ 20¢/million Btu	7,831(3)	1.37	10,417(4)	1.89
Subtotal	9,655	1.68	12,464	2.26
Capital Related @ 14% of Fixed Capital	3,696	0.65	4,438	0.80
Total	13,351	2.33(5)	16,902	3.06(6)

---

(1) 5.721×10^9 kwhr/hr
(2) 5.519×10^9 kwhr/hr
(3) 0.619 lb coal/kwhr
(4) 0.854 lb coal/kwhr
(5) 34.1 ¢/million Btu
(6) 32.4 ¢/million Btu

TABLE 38

MATERIAL BALANCE FOR PARTIAL OXIDATION OF RESIDUAL OIL

1000-Mw Nominal Size

COGAS Plant

In:

Oil:
5.88 million bbls/yr
103.8 million Btu/min

Electricity:
22 Mw
(equivalent steam)

Out:

Gas:
Temp. - 200 F
Press. - 323 psia
Flow:
589,000 ft^3/min
31,500 lb/min
HHV, 164.9 Btu/ft^3
Sensible Heat, 2.7 Btu/ft^3
Sulfur, 49 g/million Btu
Composition:

CO	25.8%
H_2	24.3%
CH_4	0.4%
H_2O	0.6%
CO_2	0.9%
N_2	48.0%
H_2S	0.01%

Sulfur 21,700 long tons/yr

Base-Load Turbine Plant

In:

Oil:
8.7 million bbl/yr
138.0 million Btu/min

Electricity:
55 Mw

Out:

Gas:
Temp. - 200 F
Press. - 529 psia
Flow:
784,000 ft^3/min
42,000 lb/min
HHV - 164.9 Btu/ft^3
Sensible Heat - 2.7 Btu/ft^3
Sulfur - 49 g/million Btu

Sulfur 29,900 long tons/yr

TABLE 39

CAPITAL COSTS FOR PARTIAL OXIDATION OF RESIDUAL OIL

1000-Mw Nominal Size Power System

	COGAS Plant Thousands of Dollars	Base-Load Gas Turbine Plant Thousands of Dollars
Gasification	700	500
Air Compression	6,600	9,300
Heat Recovery	2,100	3,100
Acid Gas Removal	1,600	1,600
Sulfur Recovery	900	1,000
Tankage	1,000	1,300
Off-sites	2,600	2,800
General Facilities	1,900	2,300
Indirects	1,500	1,800
Total Fixed Capital	18,900	23,700
$/kw	19.3	25.1

TABLE 40

ANNUAL OWNING AND OPERATING COSTS FOR PARTIAL-OXIDATION OF OIL

1000-Mw Nominal COGAS and Base-Load Turbine Systems

	COGAS			
	Thousand Dollars	Mills/kwhr(1)	Thousand Dollars	Mills/kwhr(2)
Maintenance @ 5% Fixed Capital	945	0.16	1,085	0.19
Labor @ 60,000/man/shift	1,209	0.20	1,413	0.24
Utilities				
Cooling Water @ 2¢/thousand gal	12	0.002	16	0.003
Process Water @ 30¢/thousand gal	22	0.004	29	0.005
Raw Materials				
Absorber Make-up	132	0.022	182	0.04
Sulfur Credit @ $25/long ton	(530)	(0.088)	(727)	(0.13)
Subtotal	1,790	0.30	1,998	0.35
Oil @ 30¢/million Btu	11,120	1.85	15,285	2.63
Subtotal	12,910	2.15	17,283	2.98
Capital Charges @ 14%	2,645	0.44	3,320	0.57
Total	15,555	2.59(3)	20,603	3.55(4)

---

(1) 6.01×10^9 kwhr/yr

(2) 5.8×10^9 kwhr/yr

(3) 42¢/million Btu

(4) 40.5¢/million Btu

FIG. 1

ALLOWABLE INCREASES IN STEAM SYSTEM CAPITAL COSTS FOR EQUAL POWER PRODUCTION COST

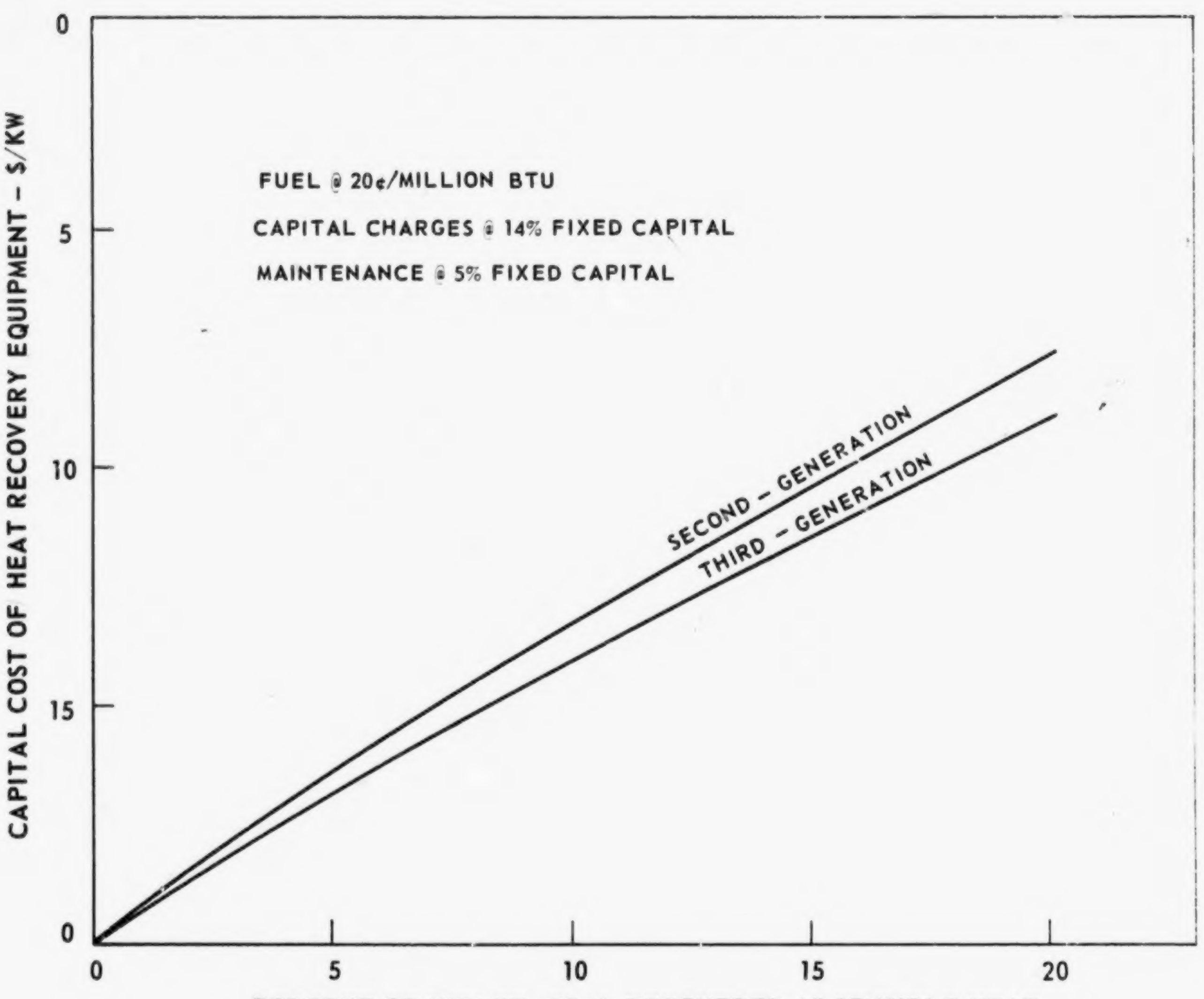
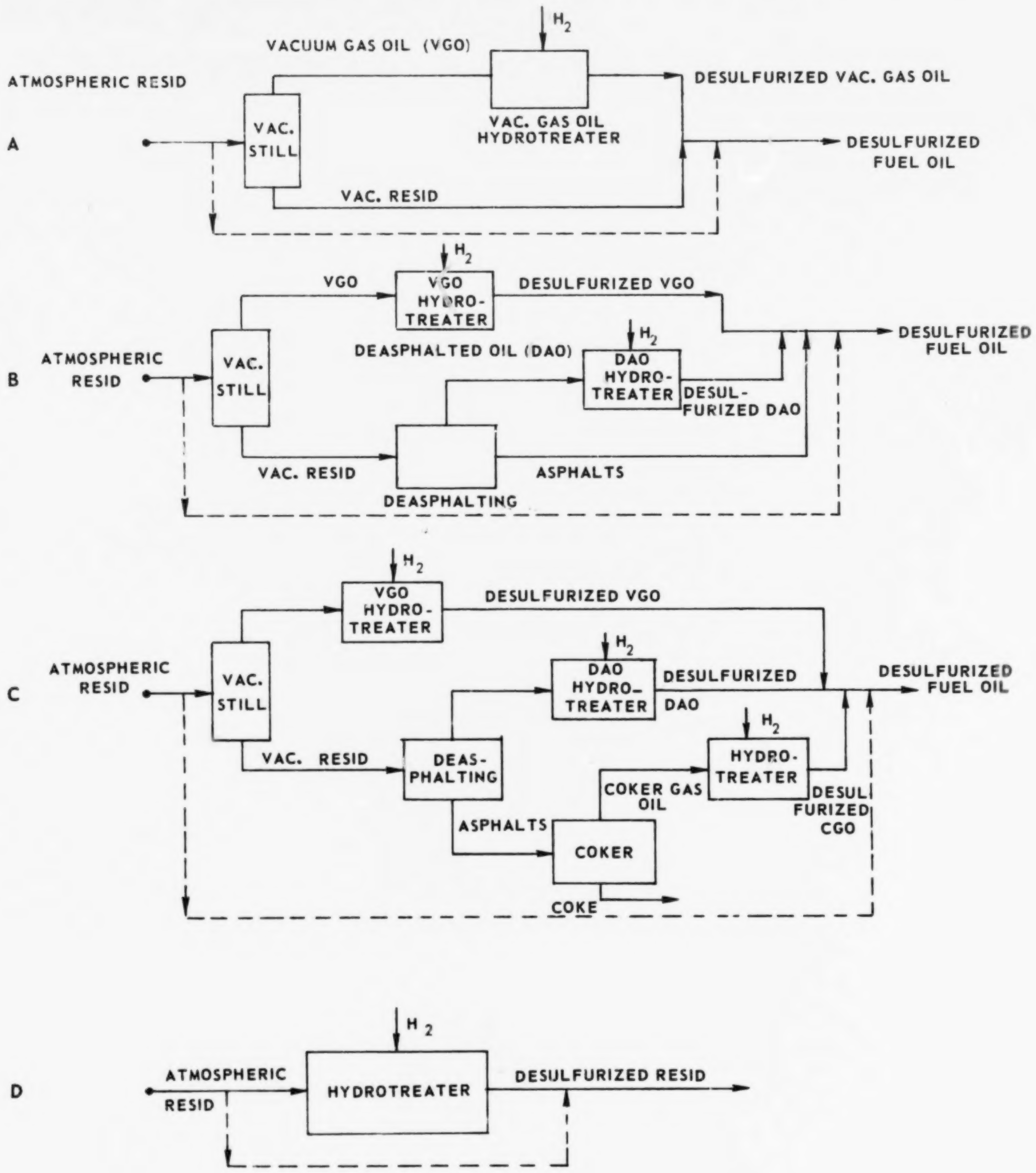


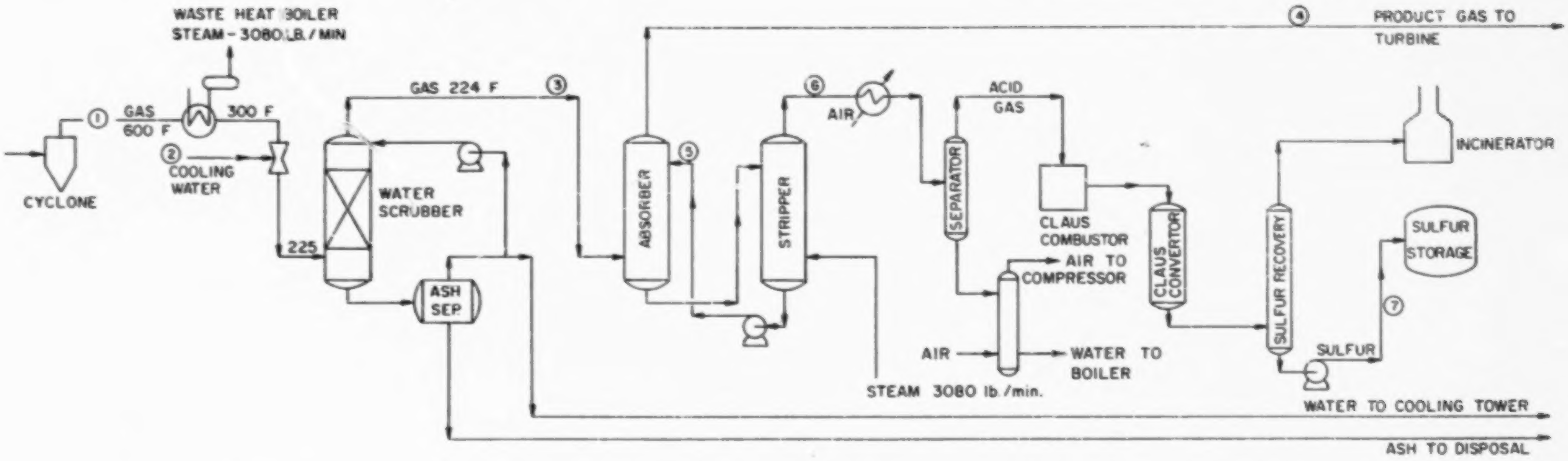
FIG. 2

HYDRODESULFURIZATION PROCESSES



HOT CARBONATE SULFUR REMOVAL SYSTEM

1000-MW NOMINAL COGAS POWER PLANT



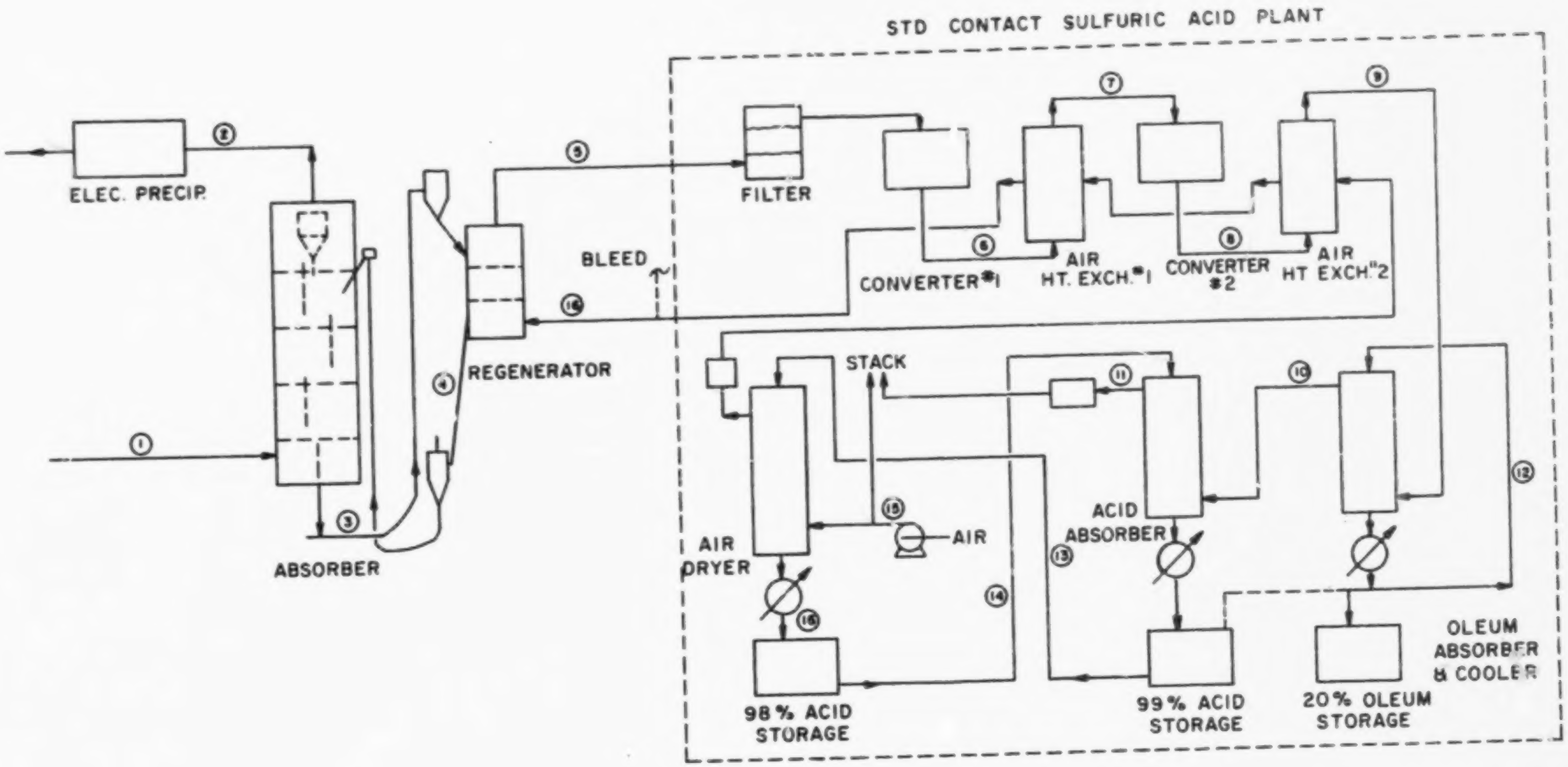
Stream No.	1		2	3		4		5	6	7	
Temp., F	600		90	224		230		230	230	260	
Pressure, psia	211		211	207		206		206	30	15	
	lb/min	lb mole/min	lb/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min
K_2CO_3								24,775			
H_2O	2,628	146.0	6,450	2,628	146.0	2,351	130.6	57,810	171.0		
H_2	778	389.0		778	389.0	778					
CO	11,845	423.0		11,845	423.0	11,845	423.0				
CO_2	2,613	59.4		2,613	59.4	700	15.2		44.5		
CH_4	128	8.0		128	8.0	128	8.0				
H_2S	312	9.2		312	9.2	5	0.14		9.36		
COS	20	0.3		20	0.3						
N_2	16,042	572.9		16,042	572.9	16,042	572.9				
Sulfur										285	8.9
Total	34,366	1607.8	6,450	34,366	1607.5	31,849	1534.8	82,585	224.86	285	8.9
ACFM or GPM	80,637		772	57,069		55,285		7,971	55,465	--	
Ave. Mol. Wt.	21.4		18	21.4		20.7		--	23.8	--	

FIG. 3

FIG. 4

HOT FERRIC OXIDE DESULFURIZATION SYSTEM

1000-MW NOMINAL SIZE COGAS POWER PLANT



Stream No.	1		2		3	4	5		16	
Temp., F	600		610		610	1000	1000		70	
Pressure, psia	195		185		185	185	185		190	
	lb /min	lb mole/ min	lb /min	lb mole/ min	lb /min	lb /min	lb /min	lb mole/ min	lb /min	lb mole/ min
H_2O	3,058	169.9	3,200	177.8						
H_2	905	452.6	910	455.2						
CO	13,782	492.2	13,782	492.2						
CO_2	3,040	69.1	3,040	69.1						
CH_4	149	9.3	149	9.3						
H_2S	364	10.7	7	0.2						
COS	24	0.4	24	0.4						
N_2	18,665	666.6	18,665	666.6			3,304	118.0	3,304	118.0
O_2	0	0	0	0			557	17.4	1,017	31.8
SO_2	-						670	10.5		
Ferric Oxide	-				2,293	2,083				
Total	39,987	1870.8	39,777	1870.4	2,293	2,083	4,531	145.9	4,321	149.8
ACFM or GPM	91,064	-	95,966	-	-	-	12,350	-	4,161	-
Avg. Mol. Wt.	21.4	-	21.3	-	-	-	31.1	-	-	-

FIG. 5

EQUILIBRIUM PARTIAL PRESSURE OF H_2S OVER CALCINED DOLOMITE

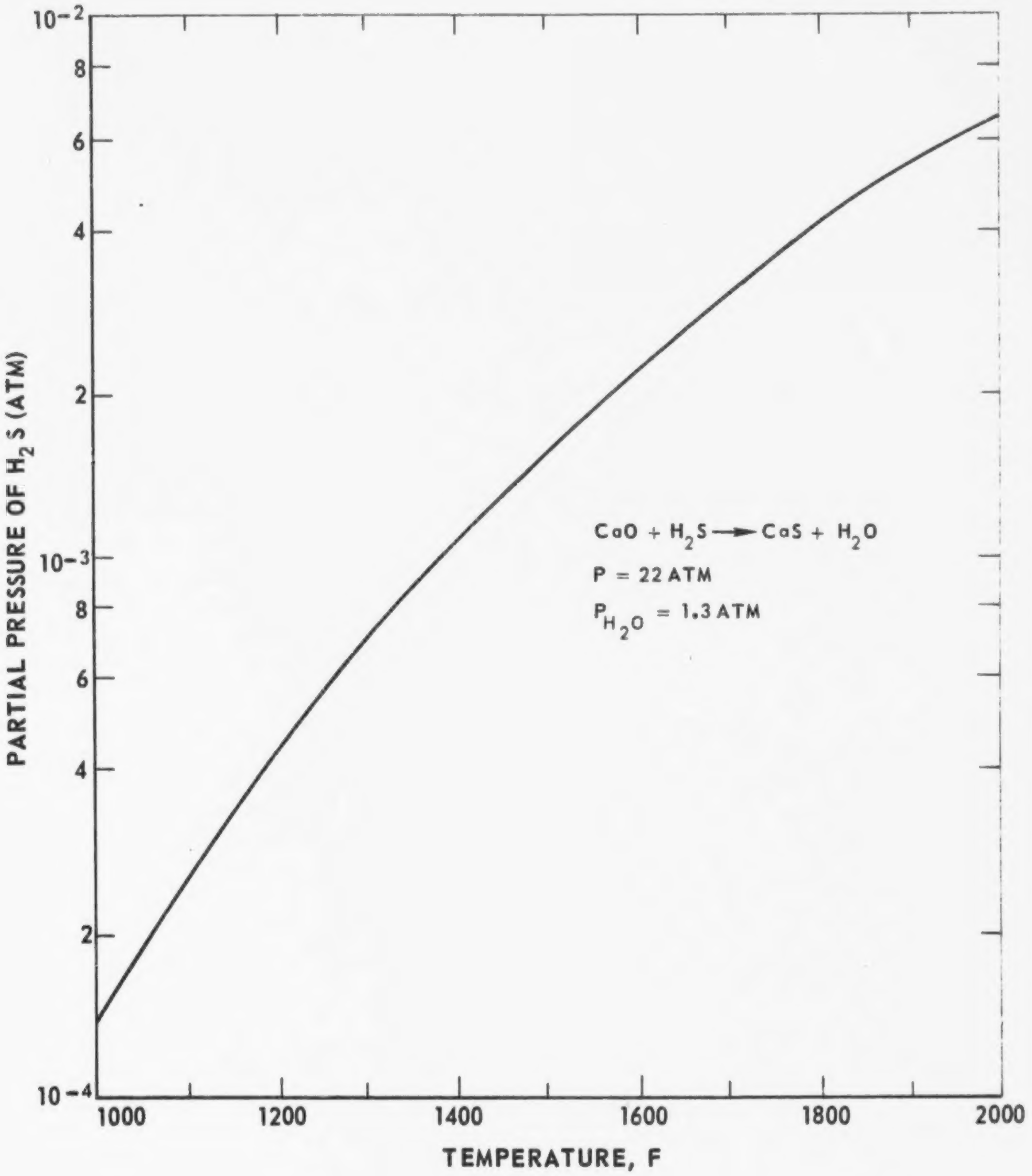


FIG. 6

EQUILIBRIUM CONCENTRATION OF H_2S IN DRY GAS AFTER REGENERATION

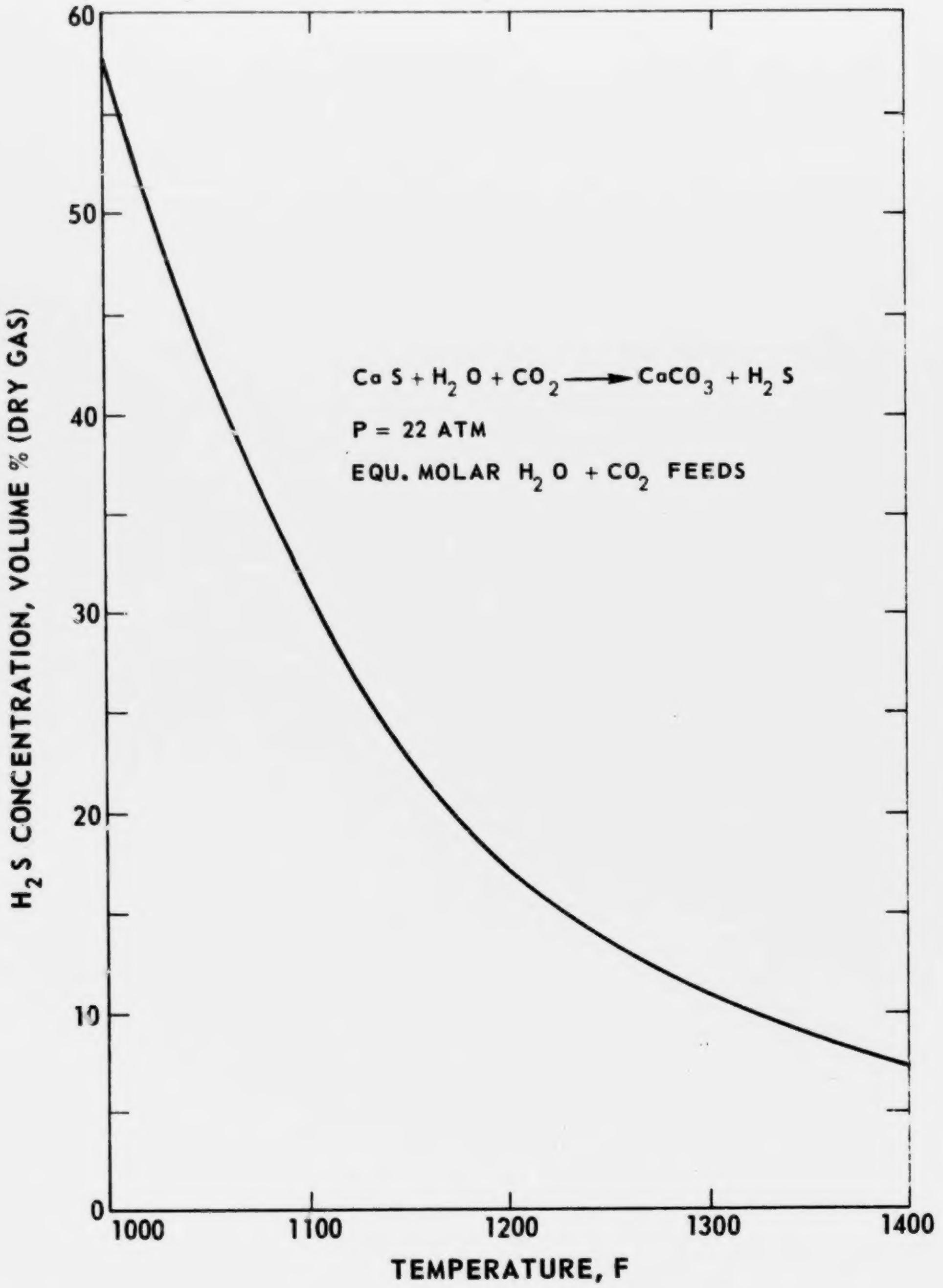
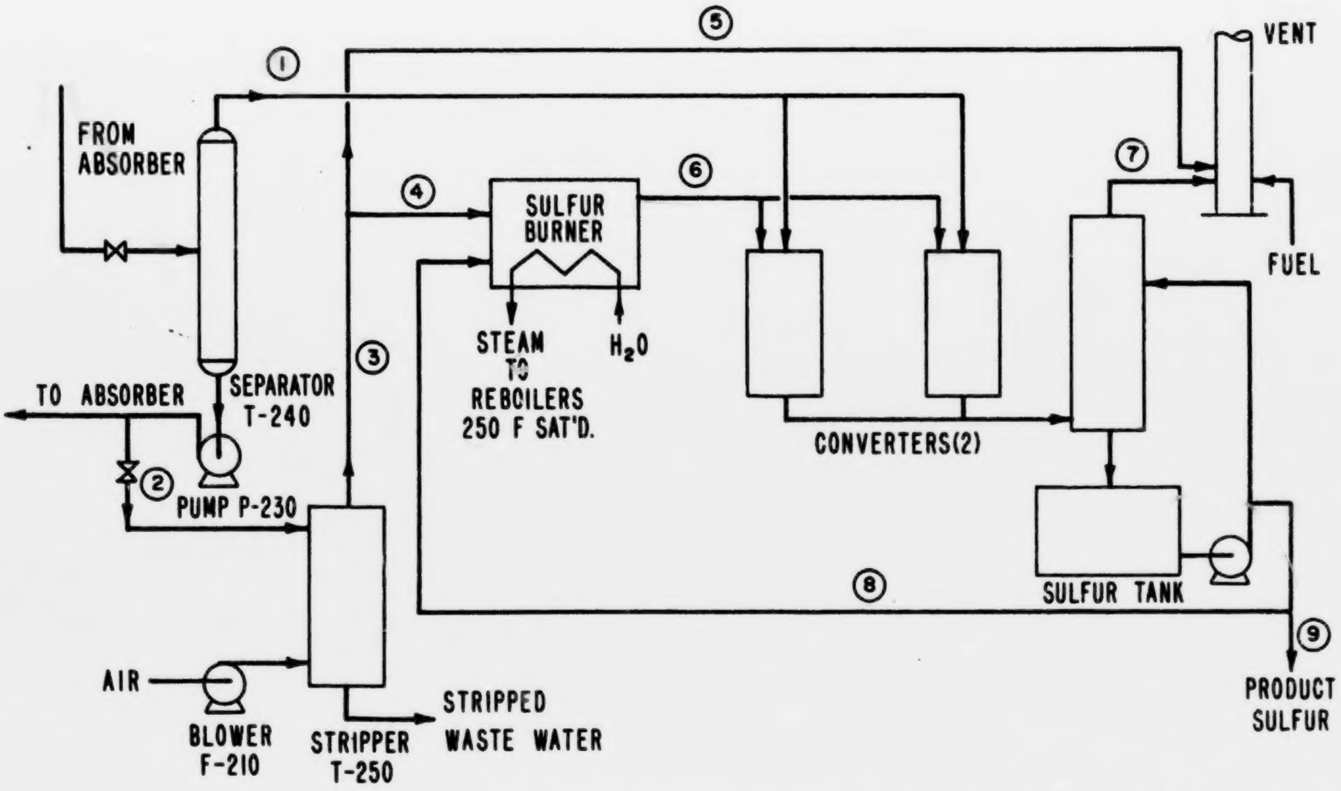


FIG. 7

CLAUS RECOVERY SYSTEM FOR LOW H_2S CONCENTRATION

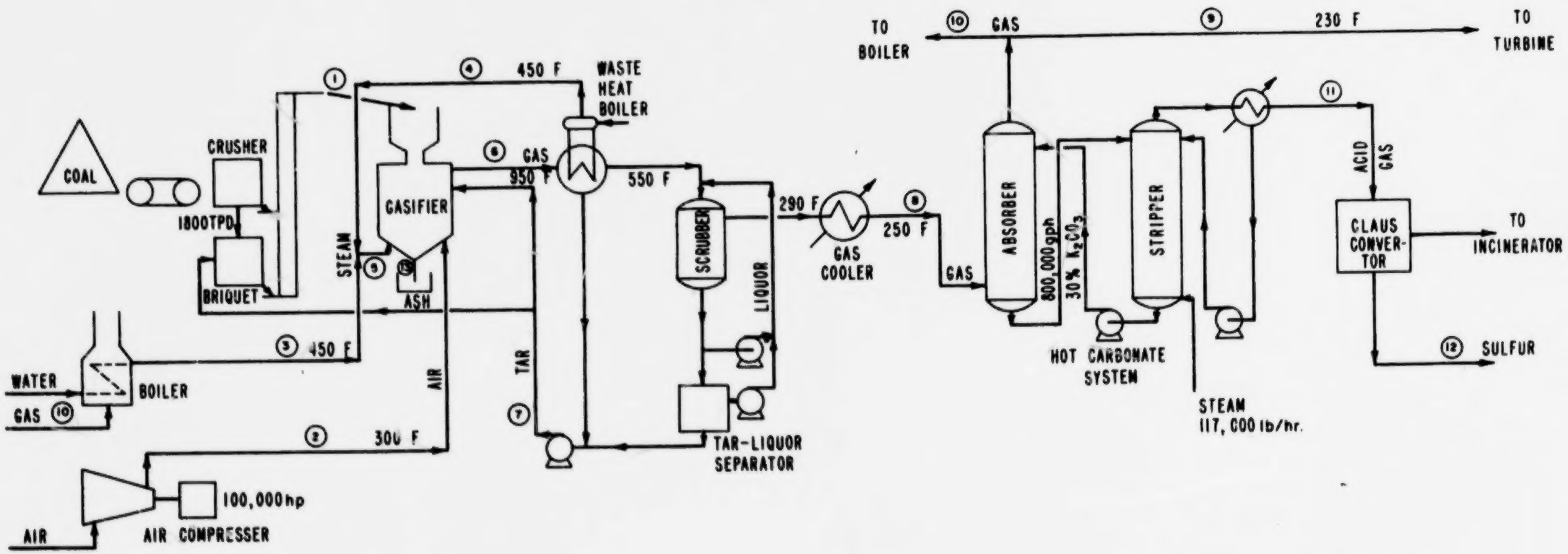


Stream No.	1	2	3	4	5	6	7	8	9
Temp F	138	138	80	80	80	757	257	257	257
Press. psia	15	15	15	15	15	15	15	15	15
H_2O lb /min	526	7,004	2,242	67	2,175	67	753		
H_2	4						4		
CO	31						31		
CO_2	12,022						12,022		
CH_4	24						24		
H_2S	328						23		
Sulfur								155	278
N_2	129		65,536	511	65,025	2,080	2,209		
O_2			19,955	155	19,800				
SO_2						1,262	43		
Total lbs/min	13,064	7,004	87,733	733	87,000	3,374	15,109	155	278

FIG. 8

LURGI GASIFICATION PROCESS

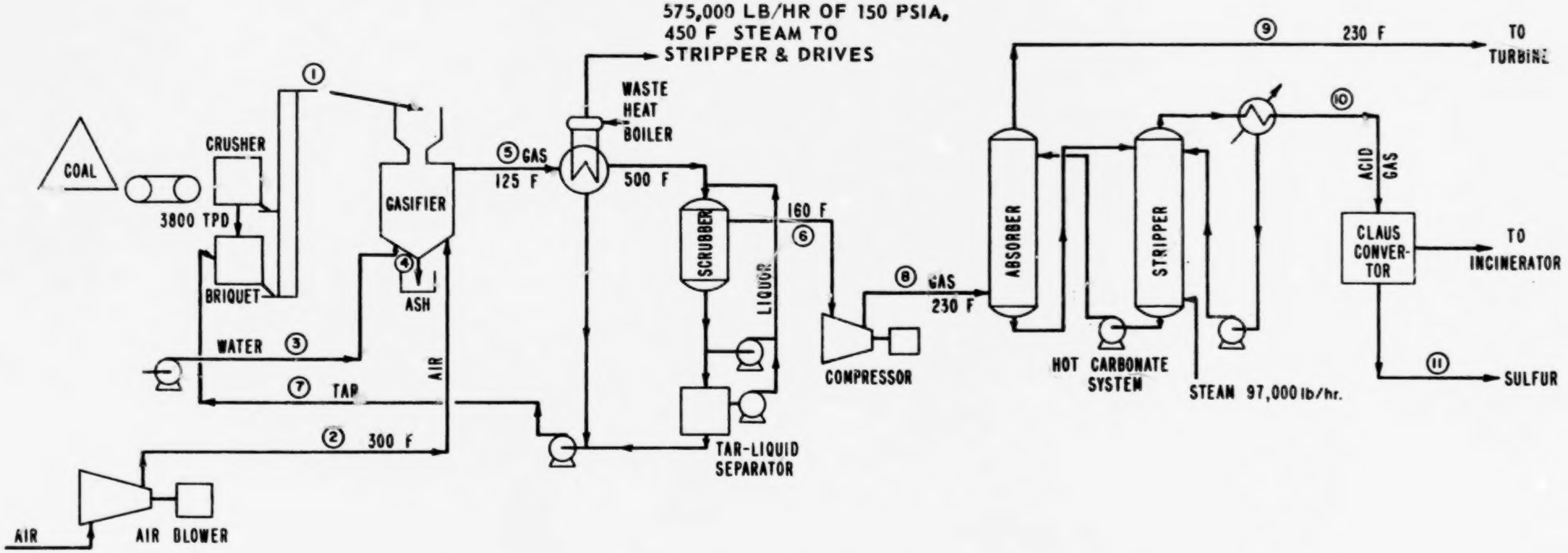
1000-MW NOMINAL COGAS POWER PLANT



Stream	(1)	(2)		(3)	(4)	(5)		(6)		(7)	(8)	(9)		(10)		(11)	(12)		(13)
Temp. F	70	300		450	450	450		950		370	250	230		230		140	250		950
Press. psia	15	325		325	325	325		322		322	317	315		315		21	15		325
	lb/min	lb/min	lb-mol/min	lb/min	lb/min	lb/min	lb-mol/min	lb/min	lb-mol/min	lb/min	lb-mol/min	lb/min	lb-mol/min	lb/min	lb-mol/min	lb-mol/min	lb/min	lb-mol/min	lb/min
H_2O	1,700			3053	4809	7862	436.8	3,645	202.5		185.6	2,144	119.1	83	4.6	5.3			
H_2								785	392.5		392.5	756	377.9	29	14.6				
CO								7,420	265.0		265.0	7,143	255.1	277	9.9				
CO_2								11,713	266.2		266.2	9,944	226.0	383	8.7	32.7			
CH_4								1,747	109.2		109.2	1,683	105.2	64	4.0				
H_2S								384	11.3		11.3	65	1.9	3	0.1	10.5			
COS								72	1.2		1.2								
N_2		20,829	743.9					20,989	749.6		749.6	20,208	721.7	781	27.9				
O_2		6,406	200.2																
Coal	11,375																		
Ash	(1380)																		1417
Tar								942		942									
Sulfur																	317	9.9	
Total	13,075	27,235	944.1	3053	4809	7862	436.8	47,697	1997.5	942	1980.6	41,943	1806.9	1620	69.8	48.5	317	9.9	1417
ACFM or GPM		23,661		4557	7177	11,734		93,820		112	47,582	42,454		1640		15,347			
Ave. Mol. Wt.		29		18	18	18		23.4			23.5	23.2		23.2		40.0			

WELLMAN-GALUSHA GASIFICATION PROCESS

1000-MW NOMINAL SIZE COGAS POWER PLANT



Stream	(1)	(2)		(3)		(4)	(5)		(6)	(7)	(8)	(9)		(10)	(11)	
Temp. F	70	85		70		1800	1250		160	400	230	230		140	250	
Press. psia	15	27		27		15	25		23	15	315	314		21	15	
	lb/min	lb/min	lb-mol/min	lb/min	lb-mol/min	lb/min	lb/min	lb-mol/min	lb-mol/min	lb/min	lb-mol/min	lb/min	lb-mol/min	lb-mol/min	lb/min	lb-mol/min
H_2O	1743			2911	161.7		3019	167.7	442.0		131.1	2320	128.9	5.4		
H_2							563	281.4	281.4		281.4	563	281.4			
CO							14729	526.8	526.8		526.8	14729	526.8			
CO_2							2714	61.7	61.7		61.7	1808	41.1	22.2		
CH_4							808	50.5	50.5		50.5	808	50.5			
H_2S							389	11.5	11.5		11.5	37	1.1	12.0		
COS							96	1.6	1.6		1.6					
N_2		25752	919.7				25863	923.7	923.7		923.7	25863	923.7			
O_2		7824	244.5													
Coal	11656															
Ash	(1167)					1715										
Tar	442						442			442						
Sulfur															346	10.8
Total	13851	33576	1164.2	2911	161.7	1715	48623	2024.9	2299.2	442	1988.3	43808	1953.5	39.6	346	10.8
ACFM or GPM		252,061		349			1,487,083		664,794	53	46,716	46,045		12,136	22	
Ave. Mol. Wt.		29		18			23.8		23.1		23.9	22.4		37.4	32	

FIG. 9

PROCESS FOR INDIRECT DESULFURIZATION OF RESIDUAL OIL

FIG. 10

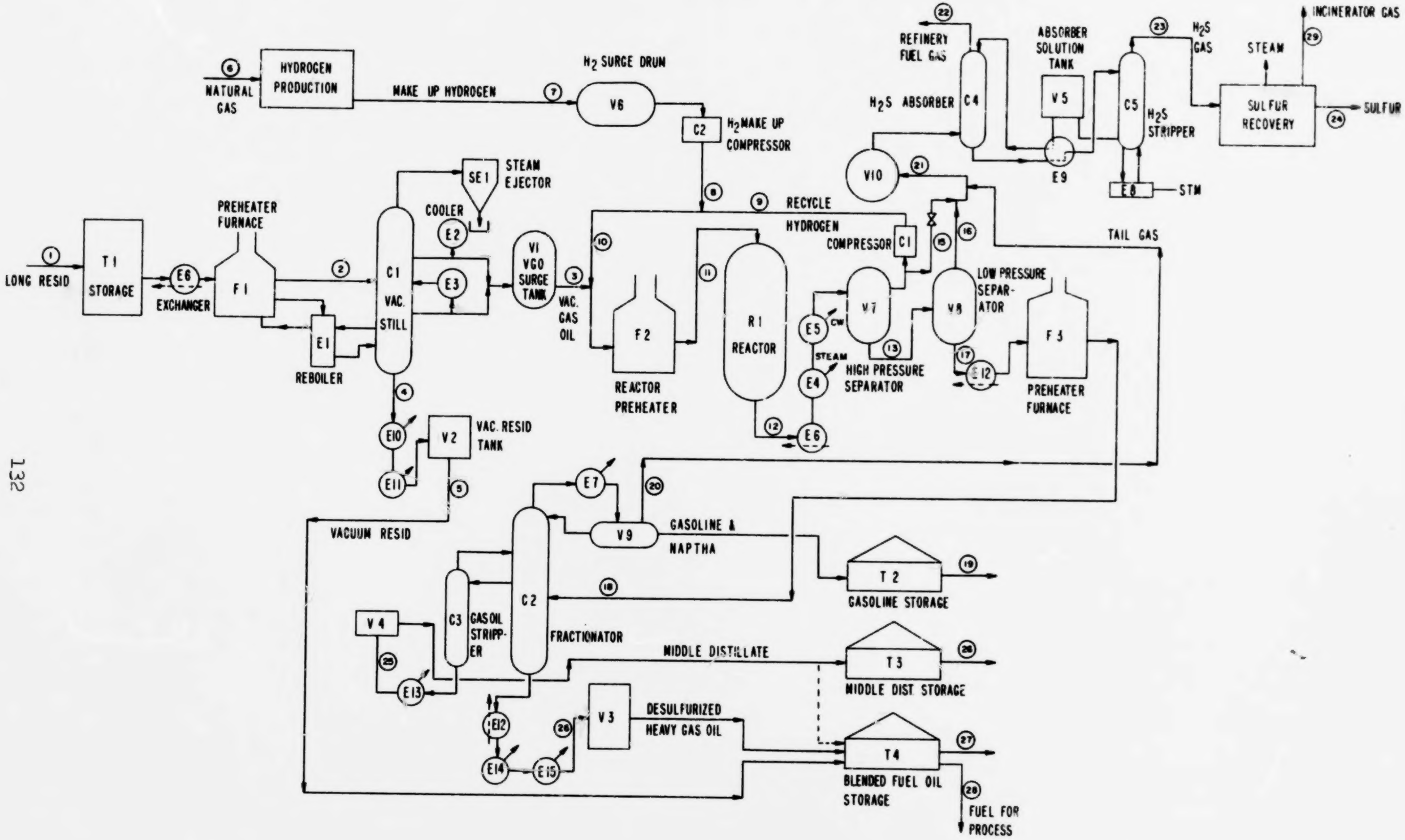
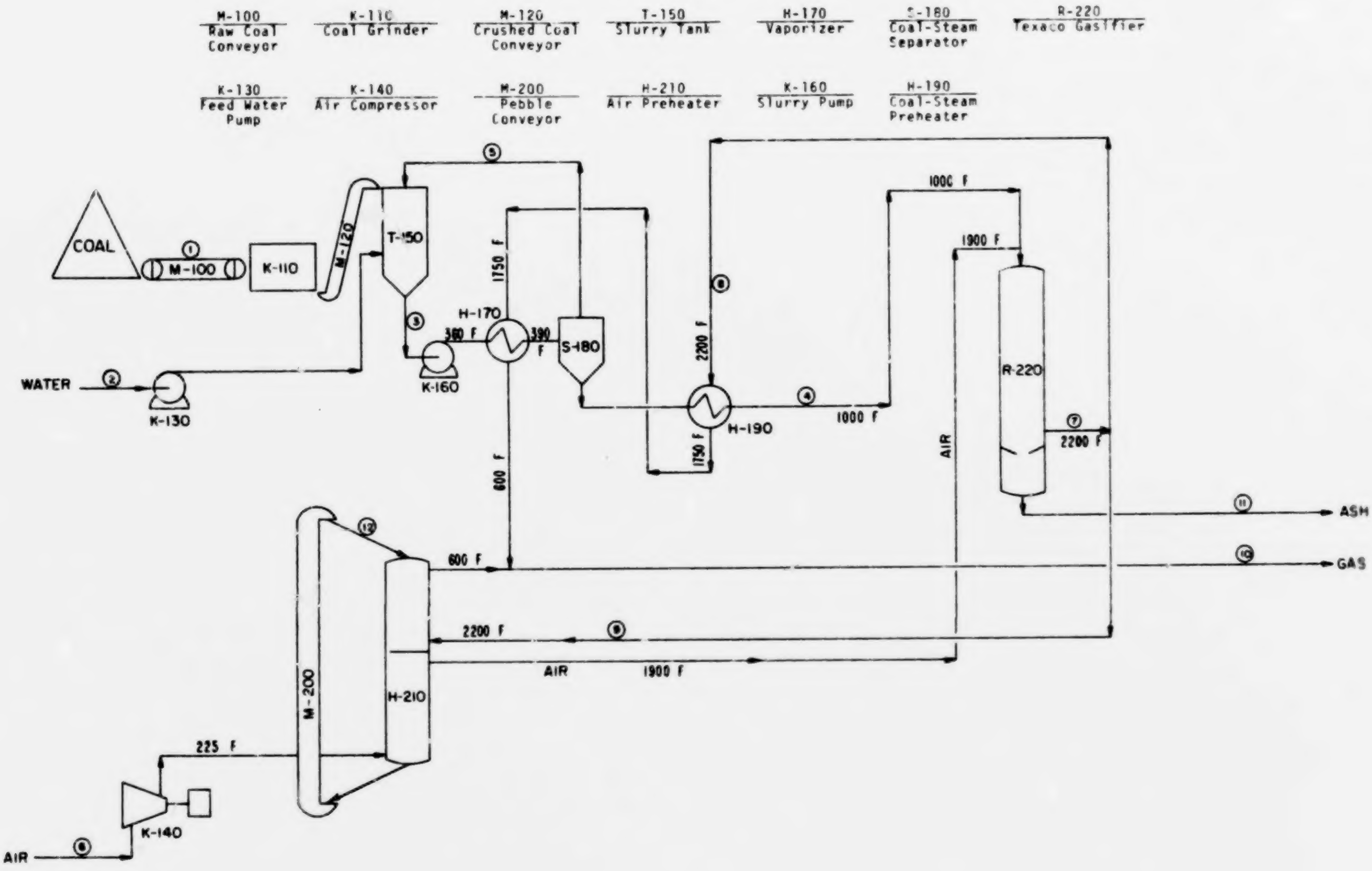


FIG. 11

SECOND-GENERATION TEXACO PARTIAL OXIDATION OF COAL PROCESS

1000-MW NOMINAL SIZE COGAS POWER PLANT

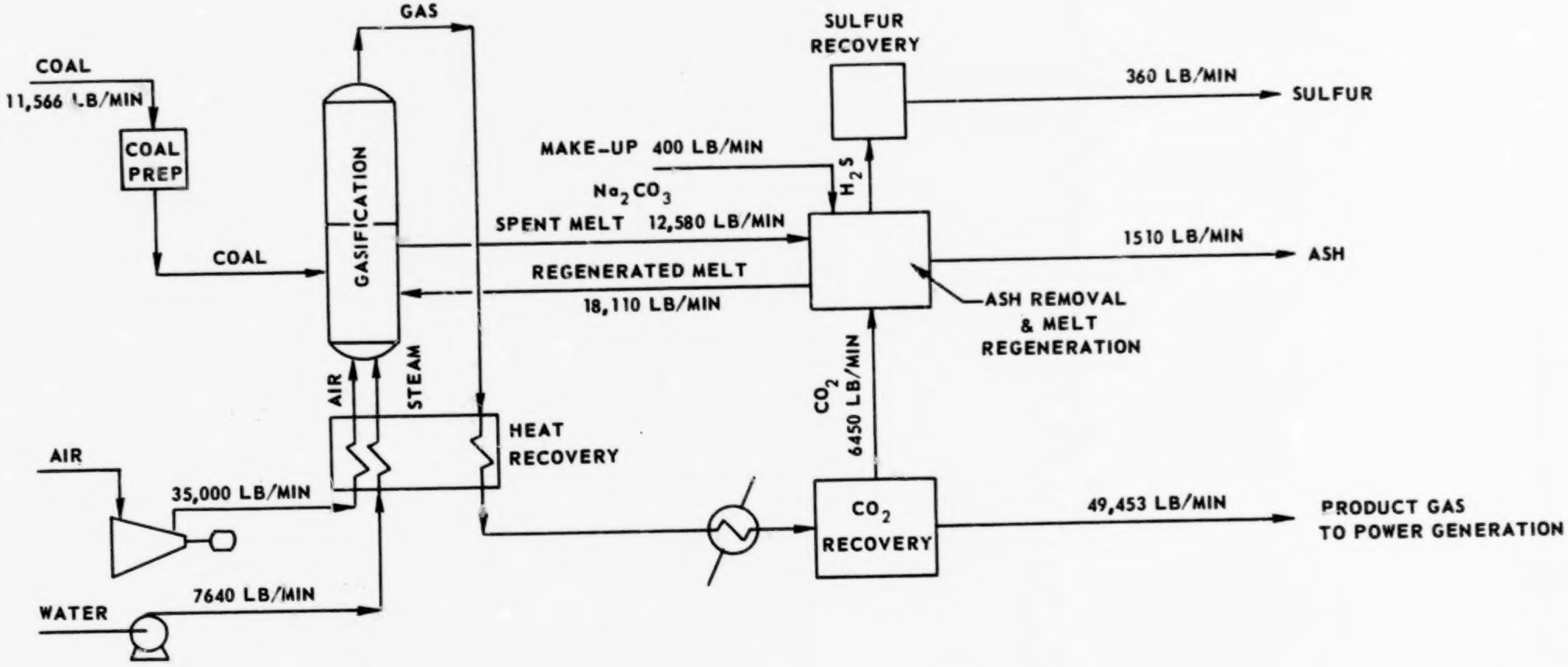


Stream No.	1	2		3		4		5		6		8	9	10		11	12
Temp., °F	70	70		360		1000		390		70		2200	2400	600		190	400
Pressure, psia	14.7	14.7		16		226		221		14.7		221	221	211		121	217
	lb/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb/min	lb/min	lb mole/min	lb/min	lb/min
Ash	(850)			(898)		(890)		(8)								1,365	
Coal	8,856			8,982		8,896		86									
H_2O	1,372	4,720	262.2	8,982	499.0	6,053	336.2	2,933	162.8			1,467	1,161	2,628	146.0		
H_2												44	--	778	389.0		
CO												6,511	5,234	11,845	423.0		
CO_2												1,459	1,154	2,617	59.4		
CH_4												31	57	128	8.0		
H_2S												174	138	312	9.2		
COS												11	9	20	0.3		
N_2										15,939	569.2	8,753	7,089	16,042	572.9		
O_2										4,843	151.3						
Alumina Pebbles																	17,200
Total	10,228	4,720	262.2	17,964	499.0	14,949	336.2	3,019	162.8	20,782	720.6	19,180	15,186	34,366	1607.8	1,365	17,200
ACFM or GPM	--	566	--	1,763	--	23,161	--	5,792	--	278,677	--	115,830	91,714	86,687	--	--	--
Ave. Mol. Wt.	--	18	--	--	--	--	--	--	--	29	--	21.4	21.4	21.4	--	--	--

FIG. 12

MODIFIED M. W. KELLOGG GASIFICATION PROCESS

1000 – Mw NOMINAL COGAS SYSTEM



PARTIAL OXIDATION OF RESIDUAL OIL

1000-MW NOMINAL SIZE POWER PLANT

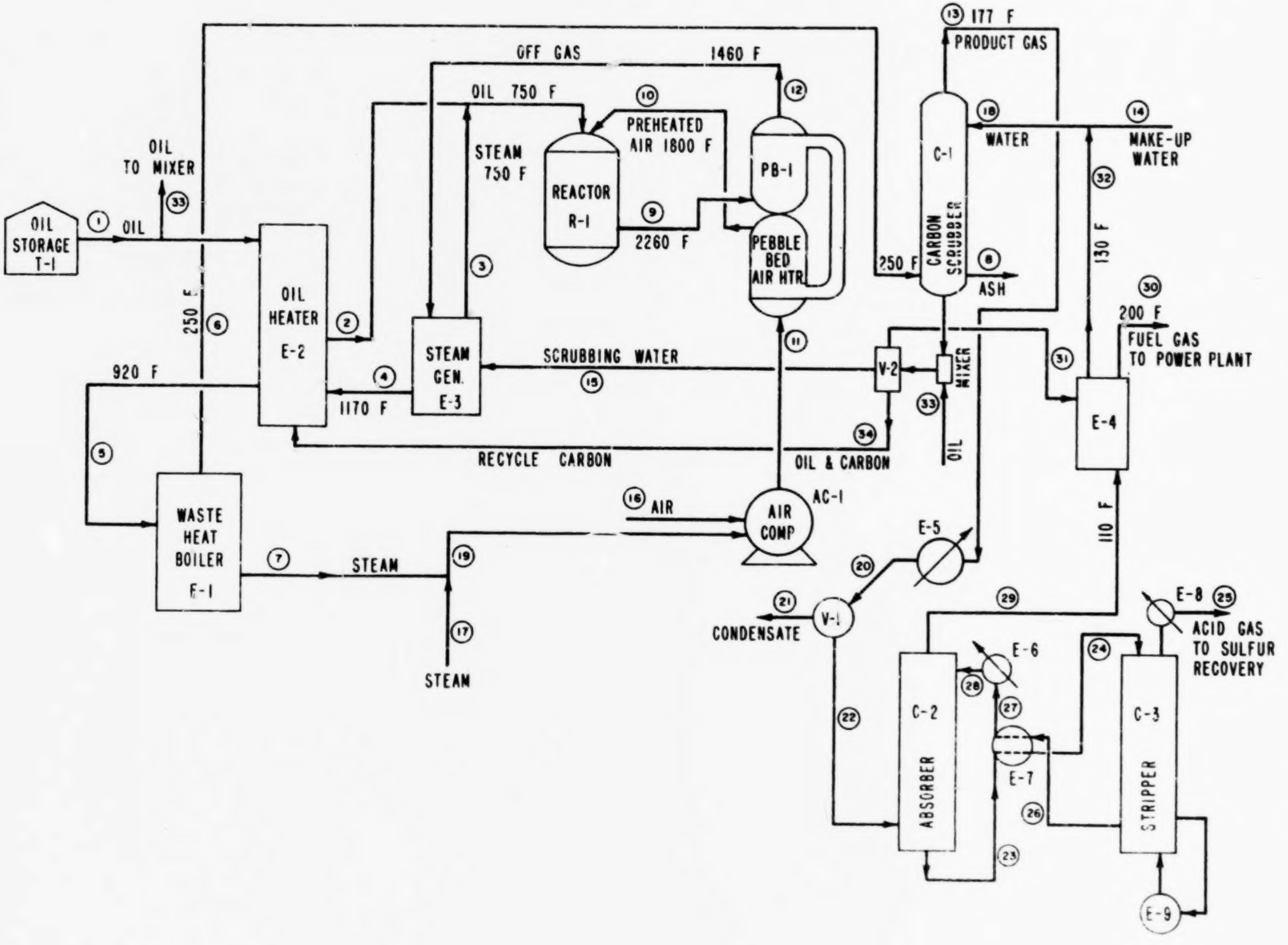


FIG. 13

FIG. 14

PROJECT "GASOLINE"- BASED PROCESS

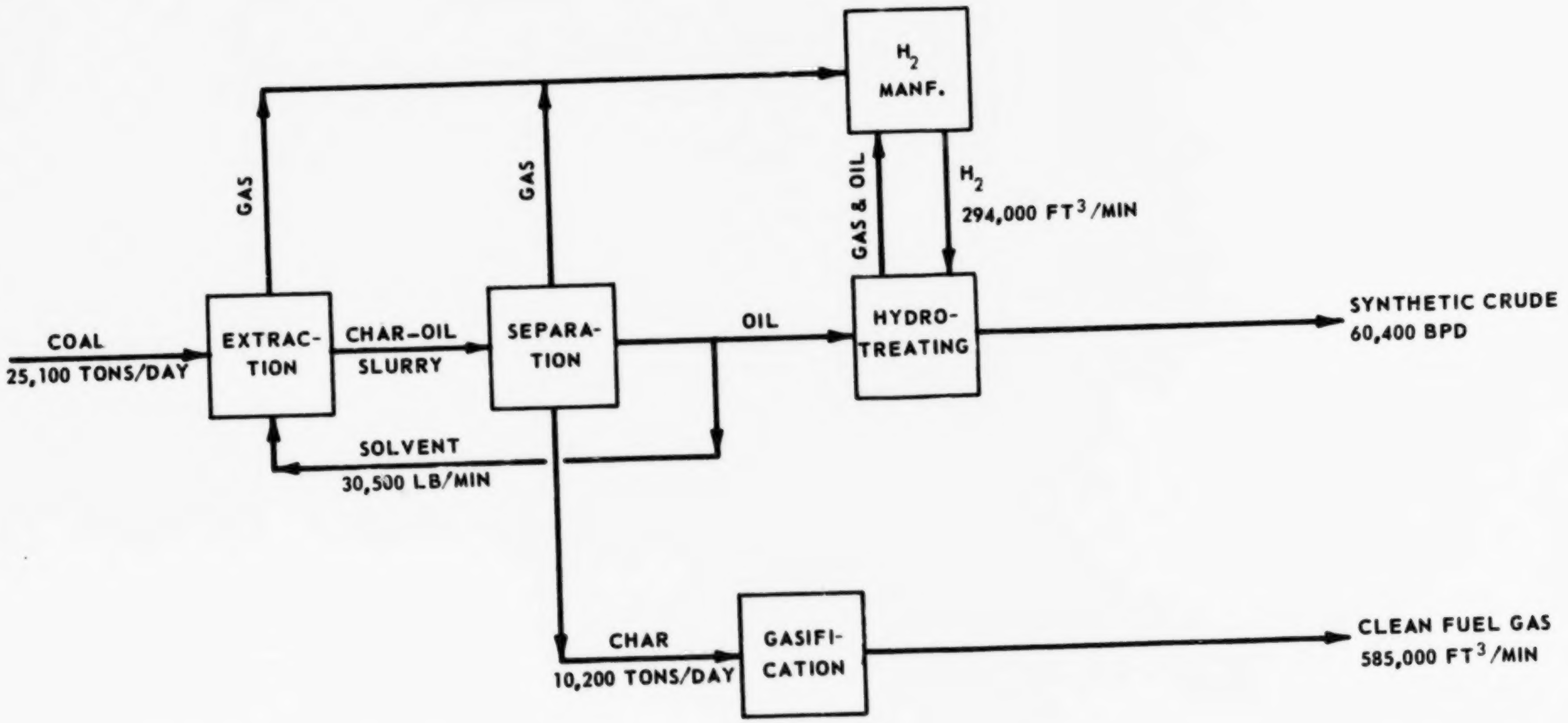


FIG. 15

EFFECT OF OIL CREDIT ON SECOND - GENERATION PROJECT 'GASOLINE' - BASED PROCESS

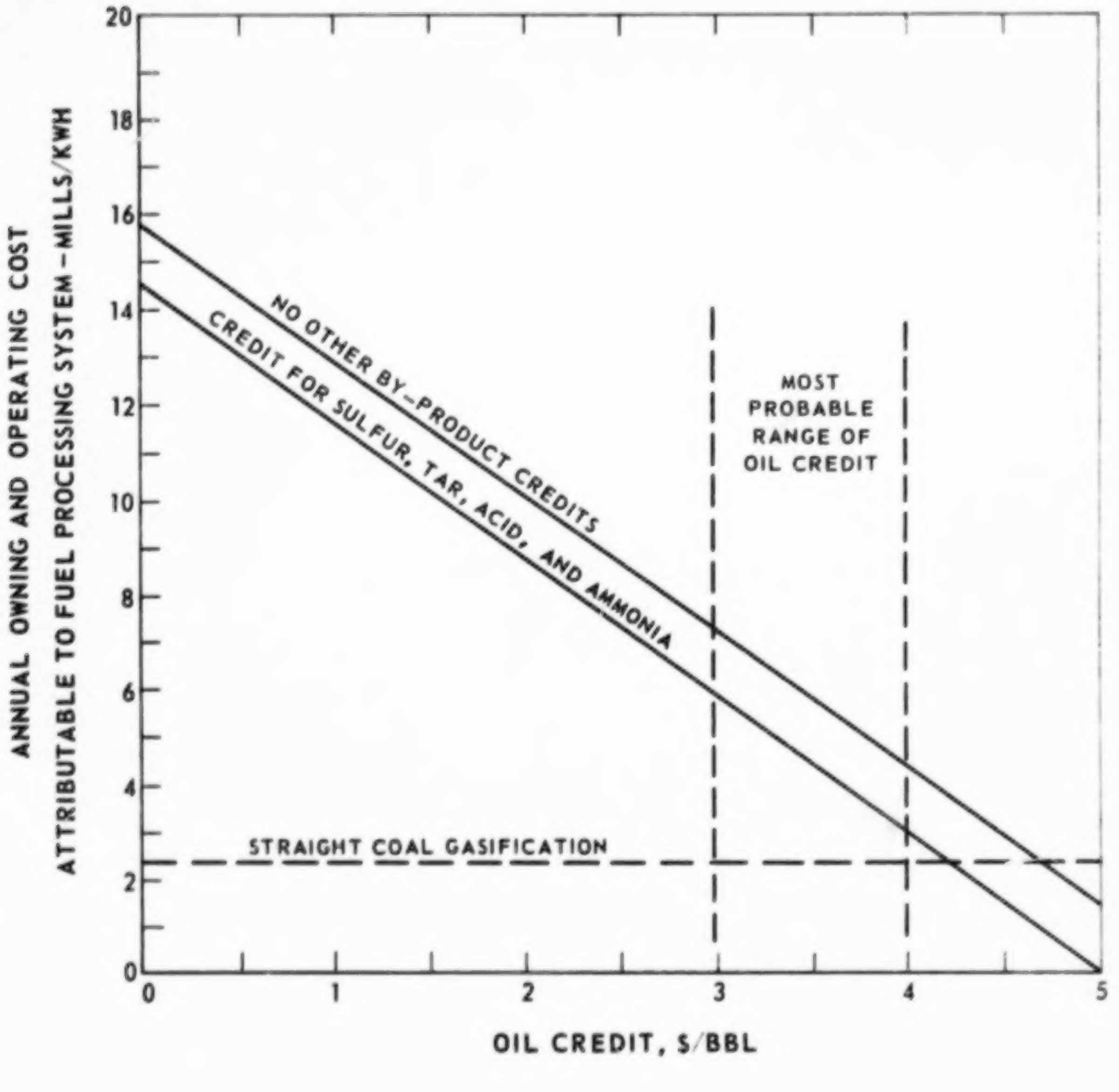
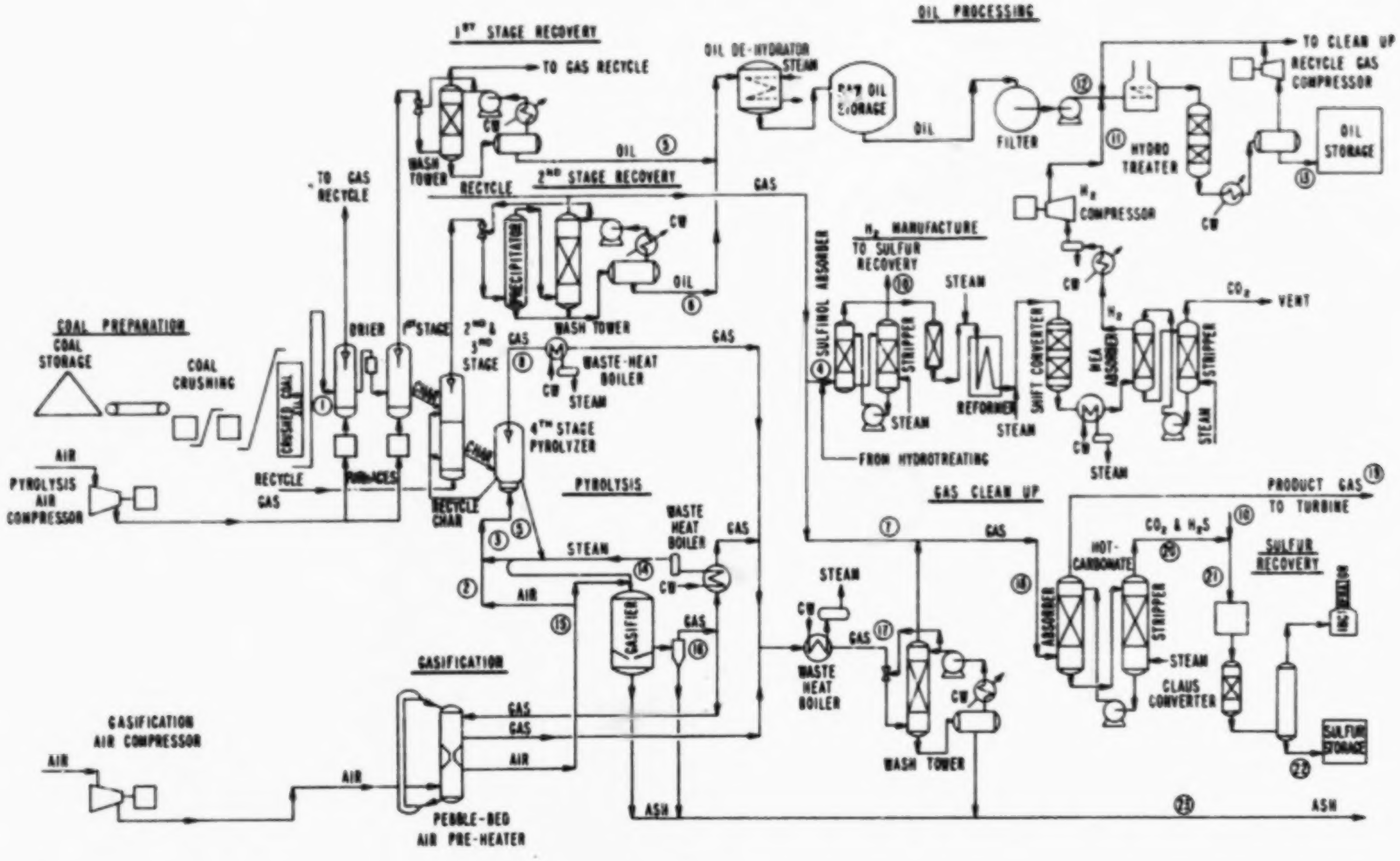


FIG. 16

COED – BASED PARTIAL OXIDATION PROCESS

1000-MW NOMINAL SIZE COGAS POWER PLANT



Stream No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Temp., F	70	1900	1000	100	250	250	100	1600	1600	200	100	250	100	1000	1900	2200	475	250	230	230	230	260	100
Pressure, psia	14.7	240	240	225	17	17	225	235	235	450	3200	3200	30	235	235	230	227	224	220	30	30	15	15
All Flows lb mole/min unless noted																							
Coal or Char	14,500*								7830*														
Oil					290*	2610*						2900*	2304*										
H_2O	2,167*		25.2					9.9			72.5			278		92.6	132.5	160.0	115.0	213.0	213.0		
H_2				2.1			6.2	21.9								205.8	227.7	233.9	233.9				
CO				1.3			3.8	66.6								354.5	425.1	428.9	429.2				
CO_2								13.2								57.4	70.6	70.6	18.0	52.6	52.6		
CH_4				9.8			29.6	4.2								7.0	11.2	40.8	40.8				
C_2^+				3.7			11.2											11.2	11.2				
H_2S				1.4			4.0	1.5		3.6						6.7	8.2	12.2	0.2	12.3	15.9		
COS																0.3	0.3	0.3	0.0				
N_2		140.7		0.2			0.8	141.4							516.5	519.7	661.1	661.9	661.9				
O_2		37.7													139.0								
Ash																							2196*
Sulfur																						15.0	
Total lb mole/min		178.4	25.2	18.5			55.6	258.7		3.6	72.5			278	655.5	1248	1506.7	1619.8	1510.2	277.9	281.5	15.0	
Total lb/min	16,667	5146	454	406	290	2610	1221	6745	7830	122	145	2900	2304	5006	18,911	29,551	36,296	38,553	35,011	6557	6689	480	2196
ACFM or GPM		18,900	1630	493			1480	24,300		57	135	327	371	18,300	71,000	155,000	66,500	54,900	50,200	68,530	72,100		
Ave. Mol. Wt.		29	18	21.9			21.9	26.1		34	2			18	29	23.7	24.2	23.7	23.1	23.5	23.7		

*Flow in lb/min.

FIG. 17

EFFECT OF OIL CREDIT ON ECONOMICS OF COED-BASED PROCESS

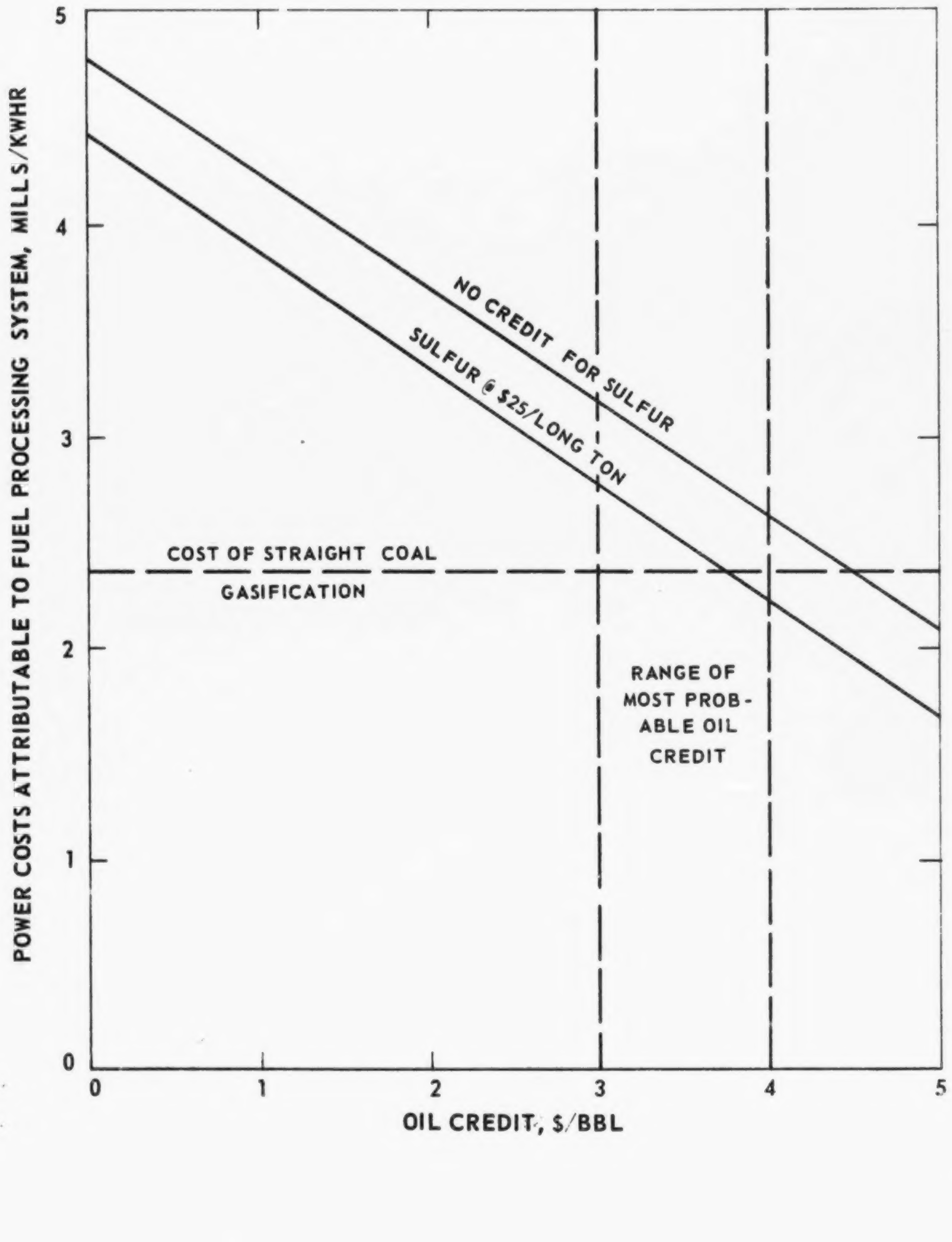
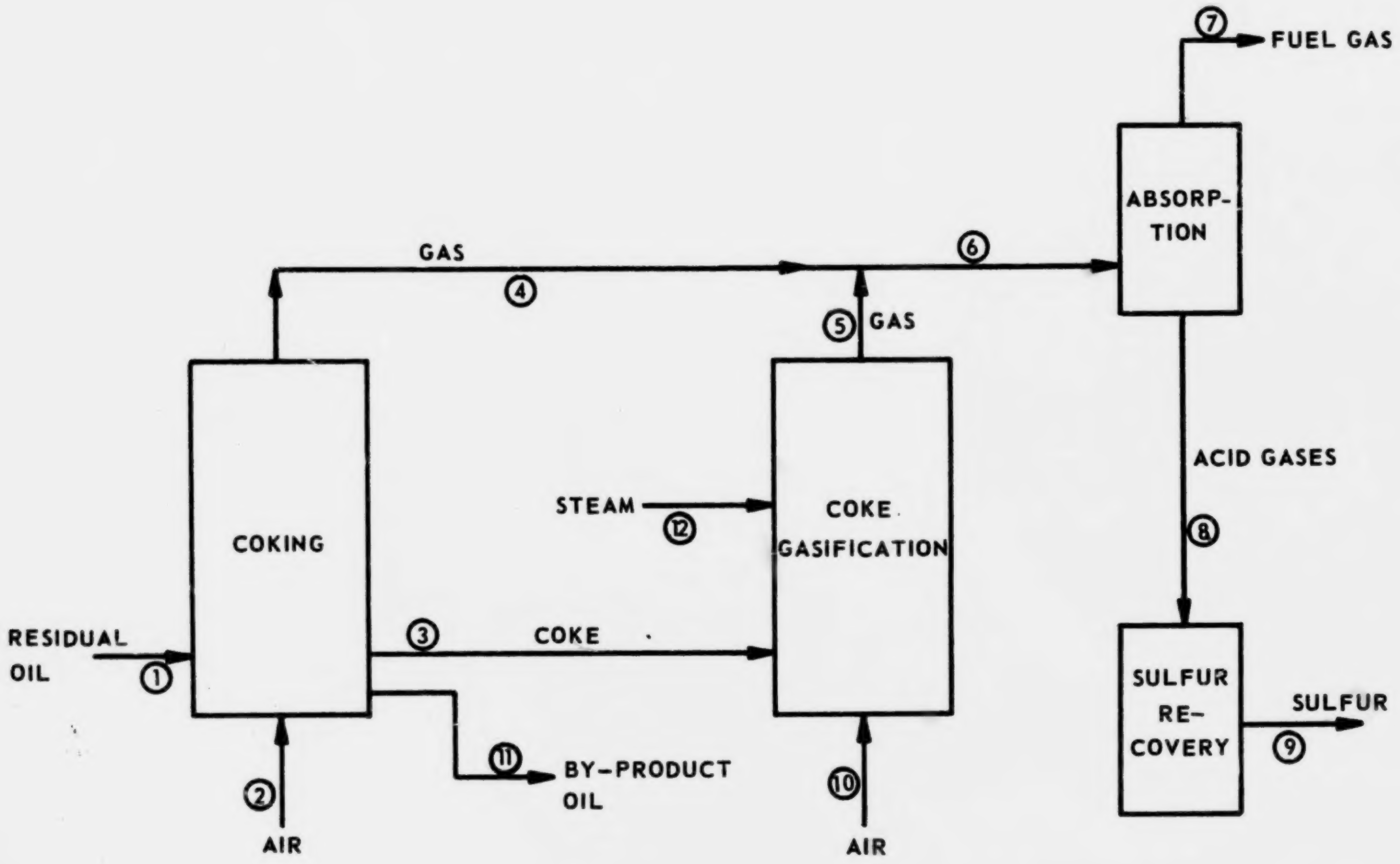


FIG. 18

FUEL GAS PRODUCTION BY COKING AND COKE GASIFICATION



Stream	1	2	3	4	5	6	7	8	9	10	11	12
Thousand lb/min	27.6	11.3	7.04	13.58	28.5	42.1	39.9	2.1	0.21	18.41	18.28	4.08
Thousand ft^3/min												
Total		141.2		215.	440.6	655.6	636.	19.97		229.5		
N_2		111.7		111.7	181.3	293.0	293.			181.3		
O_2		29.5								48.2		
H_2O				20.4	32.7	53.1	48.3	4.80				
H_2				30.6	74.3	104.9	104.9					
CO				39.15	127.0	166.2	166.2					
CO_2					20.35	20.35	7.59	12.77				
CH_4				8.6	2.4	11.0	11.0					
C_2^+				4.58		4.54	4.54					
C_2S + COS					2.49	2.49	.084	2.40				

FIG. 19

EFFECT OF BY-PRODUCT OIL VALUE ON NET COST OF CLEAN FUEL BY COKING AND COKE GASIFICATION

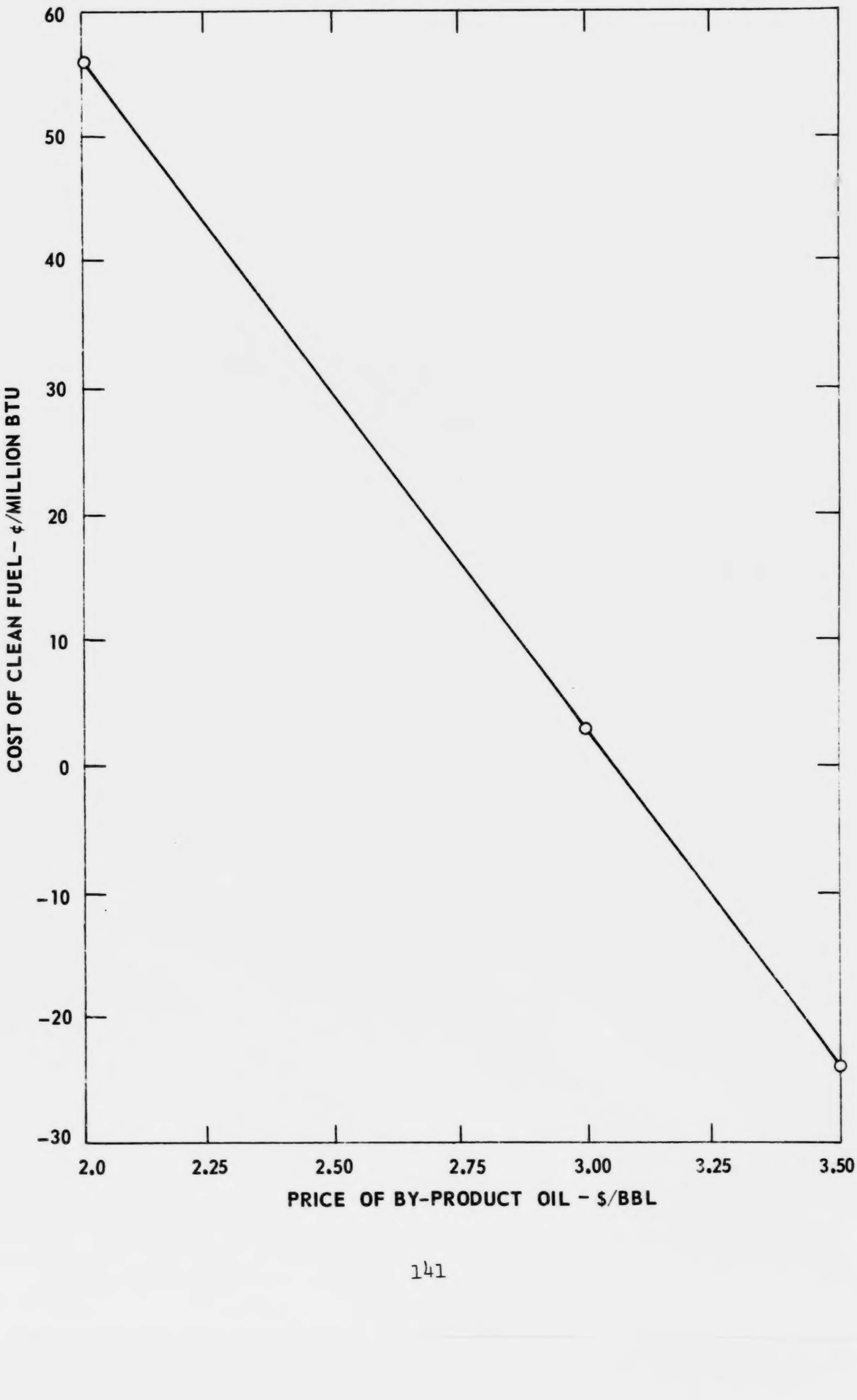
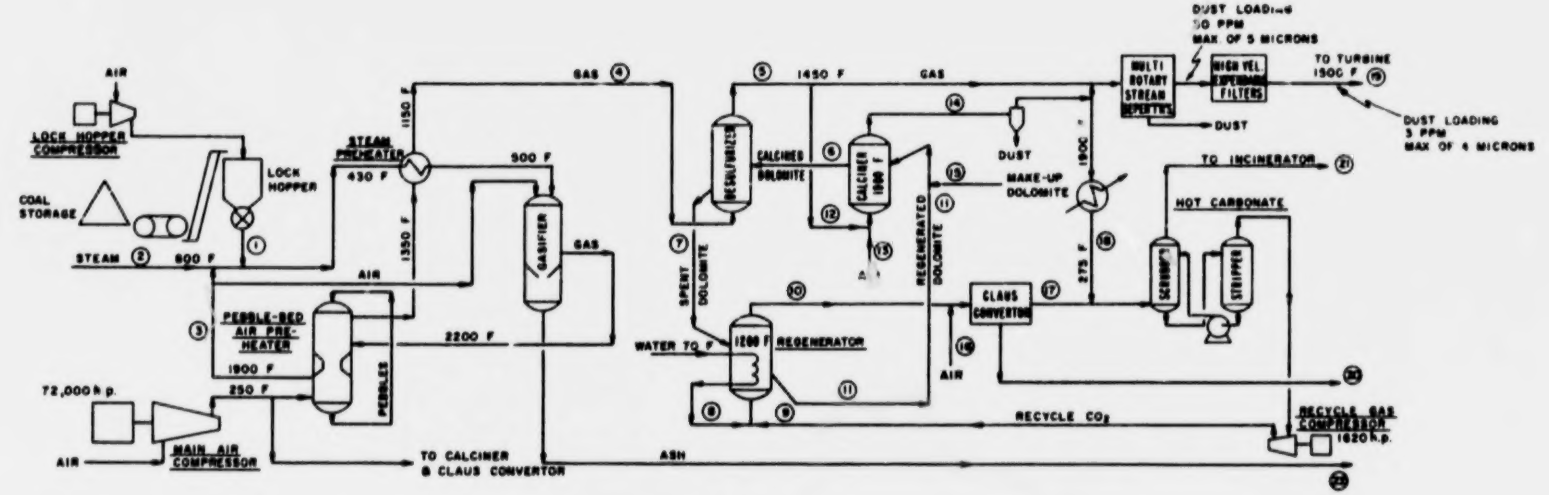


FIG. 20

NOT REPRODUCIBLE

PARTIAL OXIDATION OF COAL WITH HIGH – TEMPERATURE SULFUR REMOVAL USING DOLOMITE



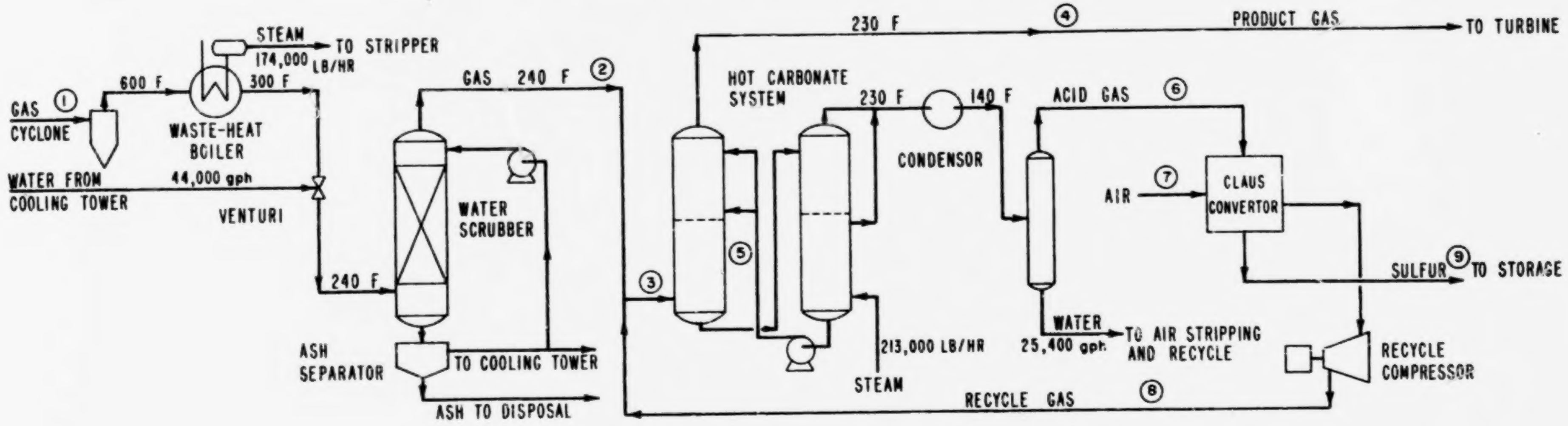
Stream No.	1	2		3		4		5	6		7	8	9	10
Temp., F.	70	900		1900		1150		1450	1900		1450	455	440	1100
Pressure, psia	350	350		350		340		335	330		335	340	340	335
	lb/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb mole/min	lb mole/min
Ash	(777)													
Coal	7,372													
H_2O	1,151	2,742	152.4			1,207	67.1	74.5				35.2	2.2	30.1
H_2						625	312.5	312.5						
CO						10,212	364.7							
CO_2						1,567	35.6	13.5					37.4	30.1
CH_4						96	6.1	6.1						
H_2S						251	7.4	0.1					0.7	8.3
COS						39	0.7	0.0						
Sulfur														
O_2				3,978	124.3									
N_2				12,929	461.7	13,014	464.7	464.6						
$CaCO_3 \cdot MgO$											22.4			
$CaO \cdot MgO$									9,520	99.2	69.2			
$CaS \cdot MgO$											7.6			
$CaCO_3 \cdot MgCO_3$														
Total	8,473	2,742	152.4	16,907	586.0	26,993	1267.4	1295.0	9,520	99.2	99.2	35.2	40.3	68.5
ACFM or CFM	--	5,720	--	42,383	--	64,373	--	85,303	--	--	--	8,110	1,144	3,421
Ave. Mol. Wt.	--	18	--	29	--	22.4	--	20.9	96	--	107	18	42.4	31.4

Stream No.	11	12	13		14	15		16		17	18	19		20		21		22
Temp., F.	1100	1450	250		1900	70		250		275	275	1500		275		250		2200
Pressure, psia	335	335	335		330	330		335		325	325	325		325		320		345
	lb mole/min	lb mole/min	lb/min	lb mole/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb mole/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min
Ash																		1,132
Coal																		
H_2O		5.3			28.4					7.6	3.5	1,692	94.0	532	29.5	40	2.2	
H_2		22.8										580	290.3					
CO		26.0										9,486	338.8					
CO_2		0.9			61.1					30.1	7.7	2,903	66.3					
CH_4		0.4										90	5.7					
H_2S		0.0								0.7		3	0.1					
COS		0.0										19	0.0					
Sulfur														245	7.6			
O_2			798	24.9				117	3.7									
N_2		33.1	2,592	92.6	125.7			381	13.6	13.6	15.1	15,192	542.2			803	28.7	
$CaCO_3 \cdot MgO$	29.7																	
$CaO \cdot MgO$	69.5				2.0													
$CaS \cdot MgO$																		
$CaCO_3 \cdot MgCO_3$						367	2.0											
Total	99.2	87.9	3,390	117.5	217.2	367	2.0	498	17.3	52.0	25.9	29,955	1337.4	775	37.1	843	30.9	1,132
ACFM or CFM	--	5,375	8,671	--	16,698	--	--	393		1,090	543	86,504	--	--	--	715	--	--
Ave. Mol. Wt.	109	20.9	29	--	31.4	184	--	29	--	35.8	31.4	22.4	--	--	--	27.3	--	--

FIG. 21

THIRD-GENERATION PARTIAL OXIDATION OF COAL WITH SULFUR REMOVAL BY HOT CARBONATE SCRUBBING

1000-MW NOMINAL COGAS POWER PLANT



Stream No.	1		2	3		4		5		6	7		8	9	
Temp., F	600		240	240		230		230		140	70		240	260	
Pressure, psia	330		326	326		325		325		22	22		326	15	
	lb/min	lb mole/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb/min	lb mole/min	lb mole/min	lb/min	lb mole/min	lb mole/min	lb/min	lb mole/min
K_2CO_3								22,112	160.0						
H_2O	2,475	137.5	84.8	1,615	90.0	1,451	80.6	51,595	2866.4	11.9			5.2		
H_2	733	366.3	366.3	733	366.3	733	366.3								
CO	11,154	398.3	398.3	11,154	398.3	11,154	398.3								
CO_2	2,461	55.9	55.9	5,482	124.6	2,461	55.9			68.7			68.7		
CH_4	121	7.5	7.5	121	7.5	121	7.5								
H_2S	324	8.7	8.7	329	9.69	2	0.05			9.94			0.33		
COS	19	0.3	0.3	19	0.3	0	0.0								
N_2	15,107	539.5	539.5	15,529	554.6	15,575	556.2				468	16.7	15.1		
O_2											130	4.5			
Sulfur														286	8.9
Total	32,364	1514.9	1461.3	34,982	1551.29	31,497	1464.85	73,707	3026.4	90.54	598	21.2	89.33	286	8.9
ACFM or GPH	32,163		33,656	35,830		33,358		7,226		26,486	5,478		2,073	--	
Ave. Mol. Wt.	21.4		21.4	22.6		21.5		--		39.5	29		39.7	32	

FIG. 22

ALLOWABLE INCREMENTAL CAPITAL COSTS FOR INCREASES IN STATION EFFICIENCY

BASE STATION – 2400 PSI/1000 F/1000 F

EFFICIENCY – 38.9%

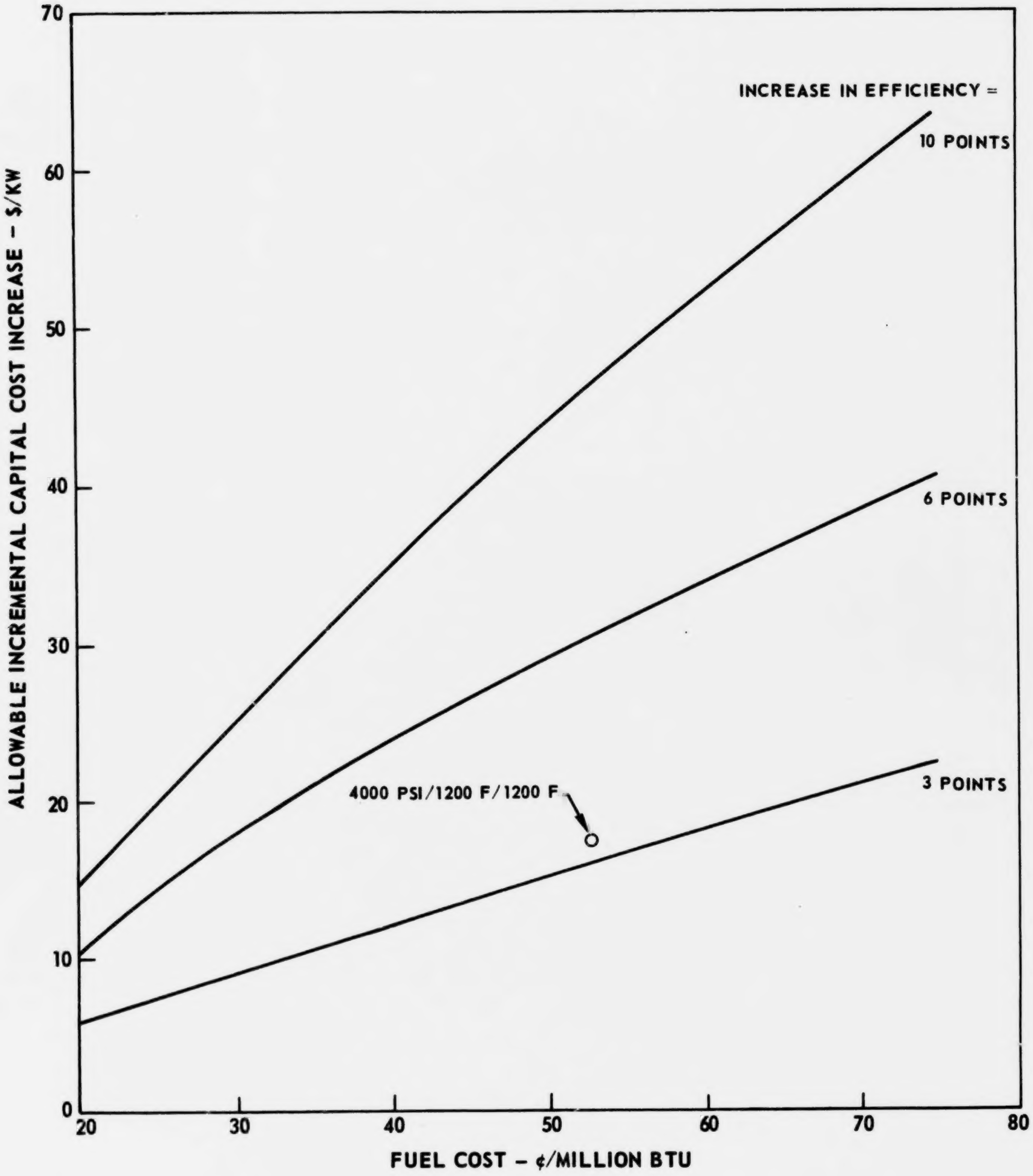
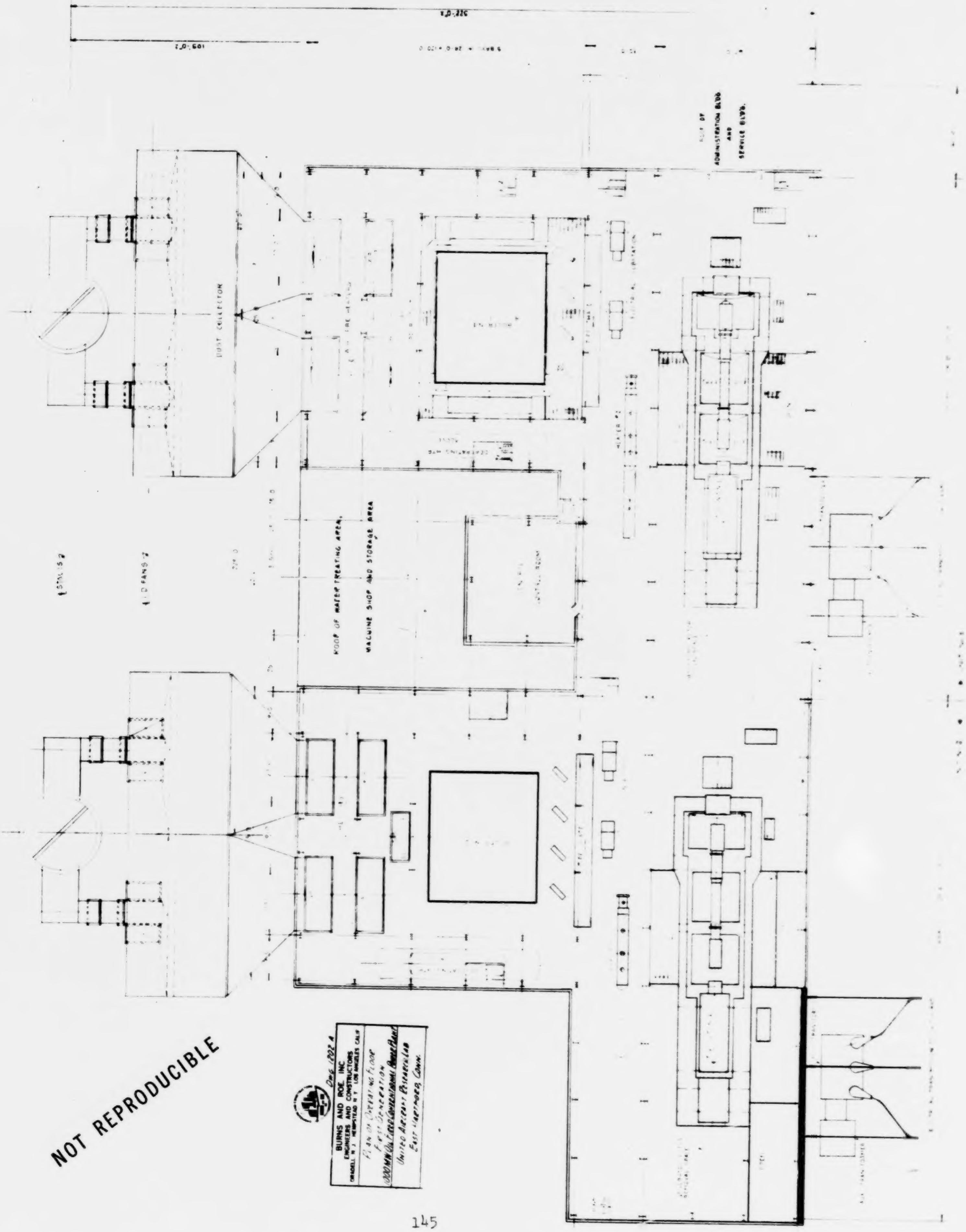
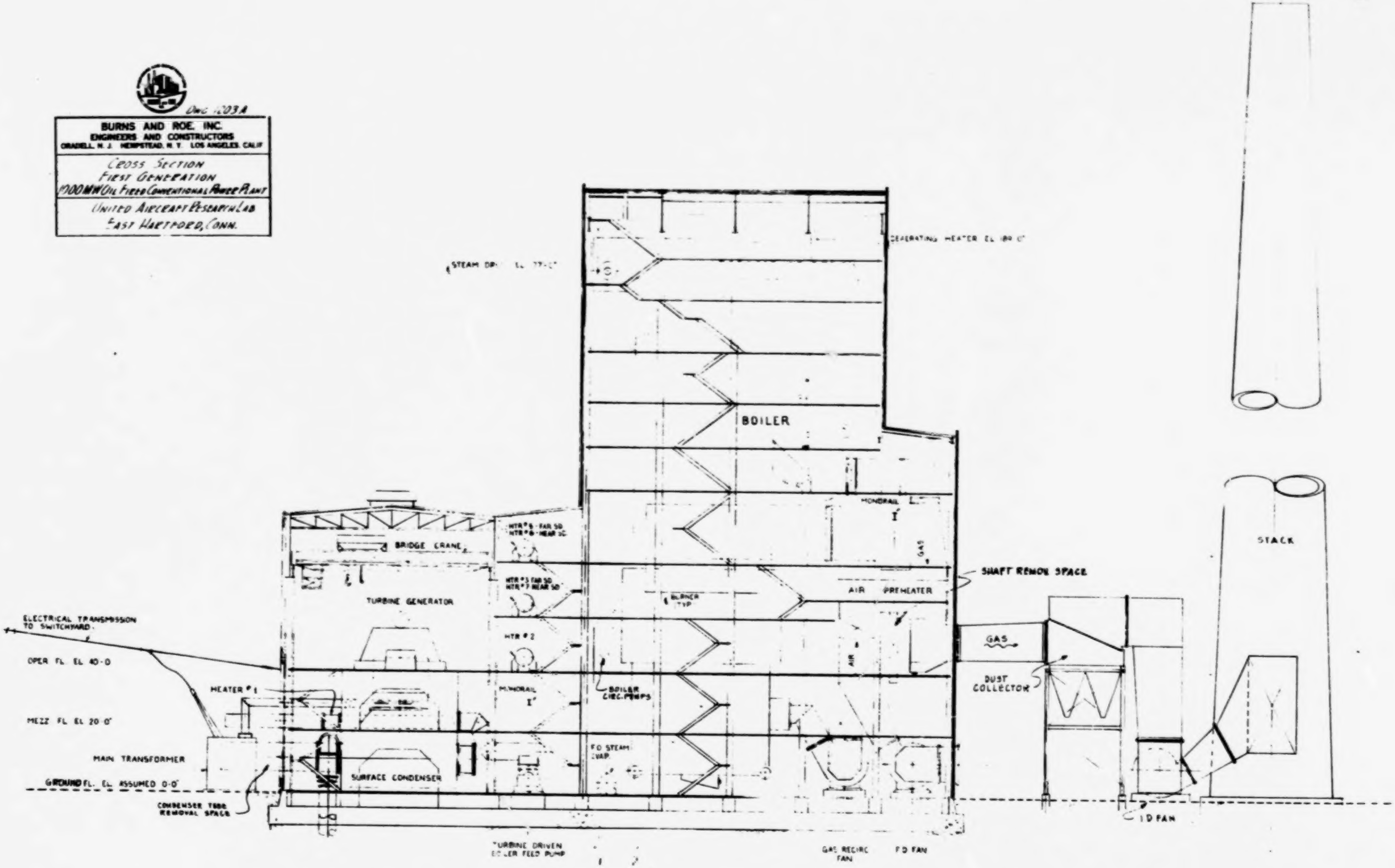


FIG. 23


DUST COLLECTOR
STACKS-2
I.D. FANS-2
ROOF OF WATER TREATING AREA, MACHINE SHOP AND STORAGE AREA
ROOF OF ADMINISTRATION BLDG AND SERVICE BLDG.
DWG 1802 A
BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL N.J. HEMPSTEAD N.Y. LOS ANGELES CALIF
PLAN OF OPERATING FLOOR
FIRST GENERATION
1000MW OIL FIRED CONVENTIONAL POWER PLANT
UNITED AIRCRAFT RESEARCH LAB
EAST HARTFORD CONN.

NOT REPRODUCIBLE

FIG. 24


Dwg 1203A
BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL, N. J. HEMPSTEAD, N. Y. LOS ANGELES, CALIF
Cross Section
First Generation
1300 MW Oil Fired Conventional Power Plant
United Aircraft Research Lab
East Hartford, Conn.
℄ STEAM DR. EL 77-0"
DEAERATING HEATER EL 189-0"
BOILER
STACK
HTR #5 - FAR SD
HTR #6 - NEAR SD
HTR #3 FAR SD
HTR #7 NEAR SD
HTR #2
MONORAIL
MONORAIL
BRIDGE CRANE
TURBINE GENERATOR
BURNER TYP
GAS
AIR PREHEATER
AIR
SHAFT REMOVAL SPACE
GAS
DUST COLLECTOR
ELECTRICAL TRANSMISSION TO SWITCHYARD
OPER FL EL 40-0
MEZZ FL EL 20-0"
HEATER #1
MAIN TRANSFORMER
GROUND FL. EL. ASSUMED 0-0"
CONDENSER TUBE REMOVAL SPACE
SURFACE CONDENSER
BOILER CIRC. PUMPS
F.D. STEAM EVAP.
TURBINE DRIVEN BOILER FEED PUMP
GAS RECIRC FAN
F.D FAN
I.D FAN

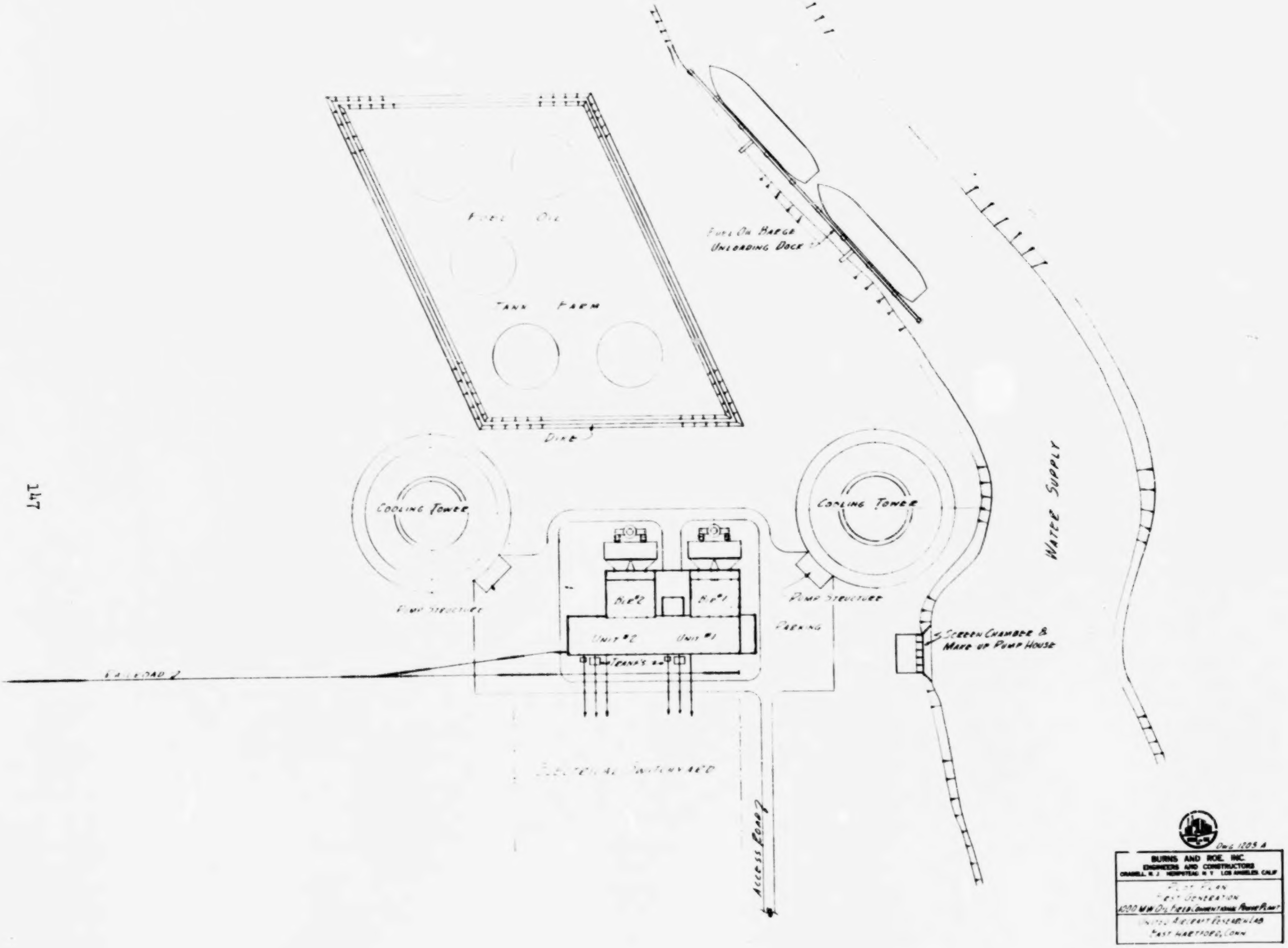
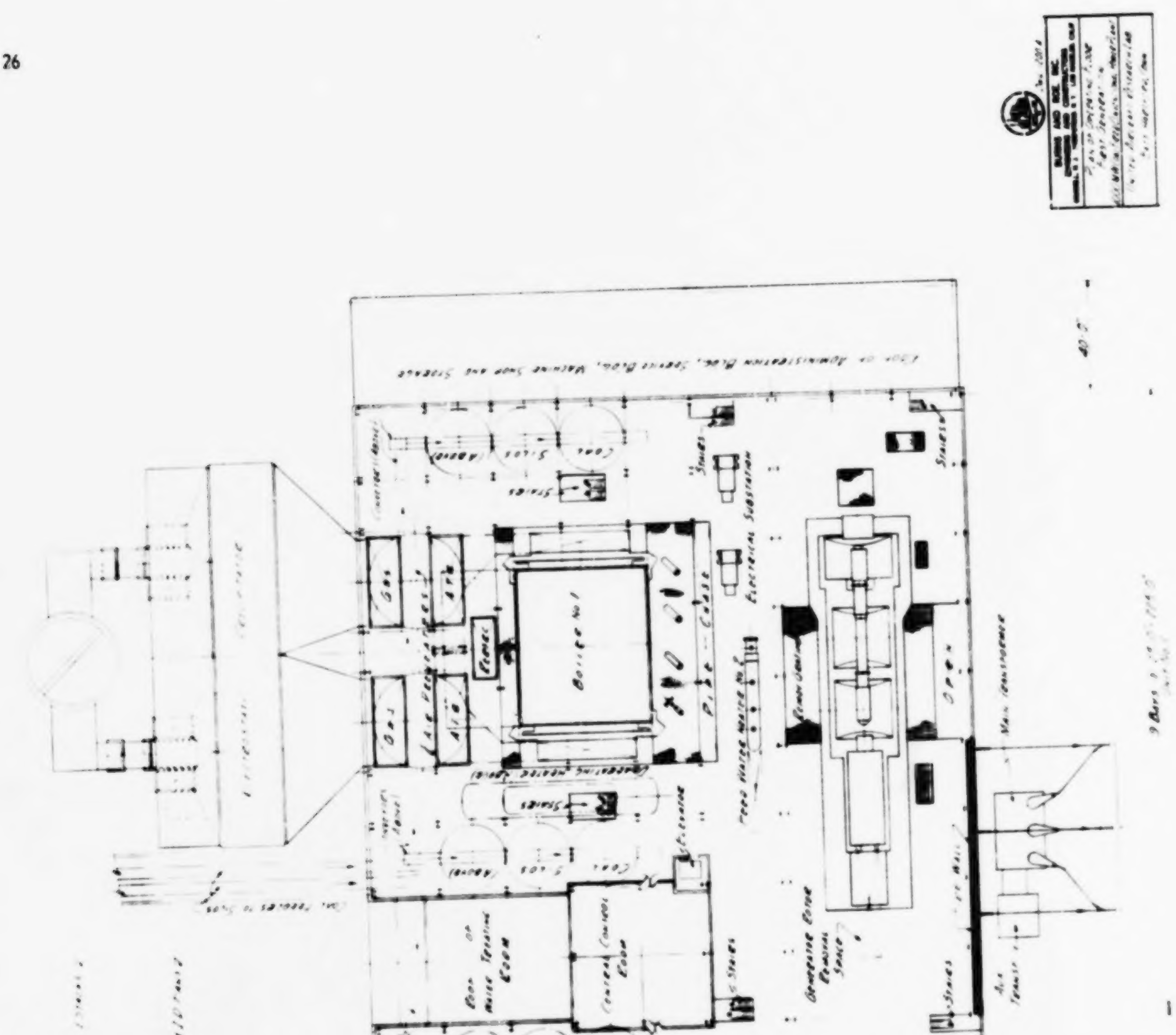
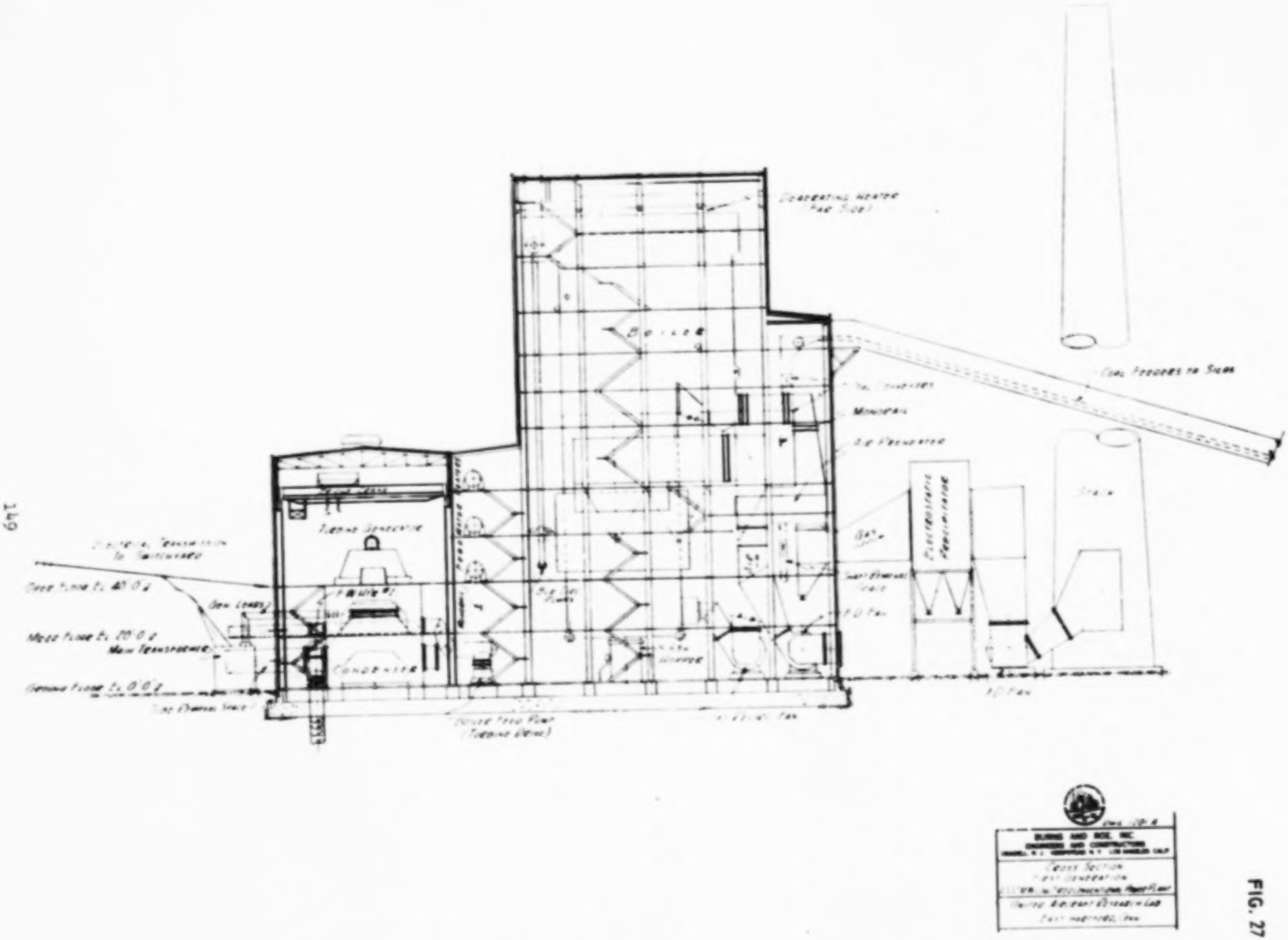

FUEL OIL
TANK FARM
DIKE
FUEL OIL BARGE UNLOADING DOCK
WATER SUPPLY
COOLING TOWER
COOLING TOWER
PUMP STRUCTURE
PUMP STRUCTURE
PARKING
SCREEN CHAMBER & MAKE UP PUMP HOUSE
BLR #2
BLR #1
UNIT #2
UNIT #1
TRANS'S
RAILROAD
ELECTRICAL SWITCHYARD
ACCESS ROAD
DWG 1205 A
BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL, N.J. HEMPSTEAD, N.Y. LOS ANGELES, CALIF
PLOT PLAN
FIRST GENERATION
1000 MW OIL FIRED CONVENTIONAL POWER PLANT
UNITED AIRCRAFT RESEARCH LAB
EAST HARTFORD, CONN

FIG. 25

FIG. 26

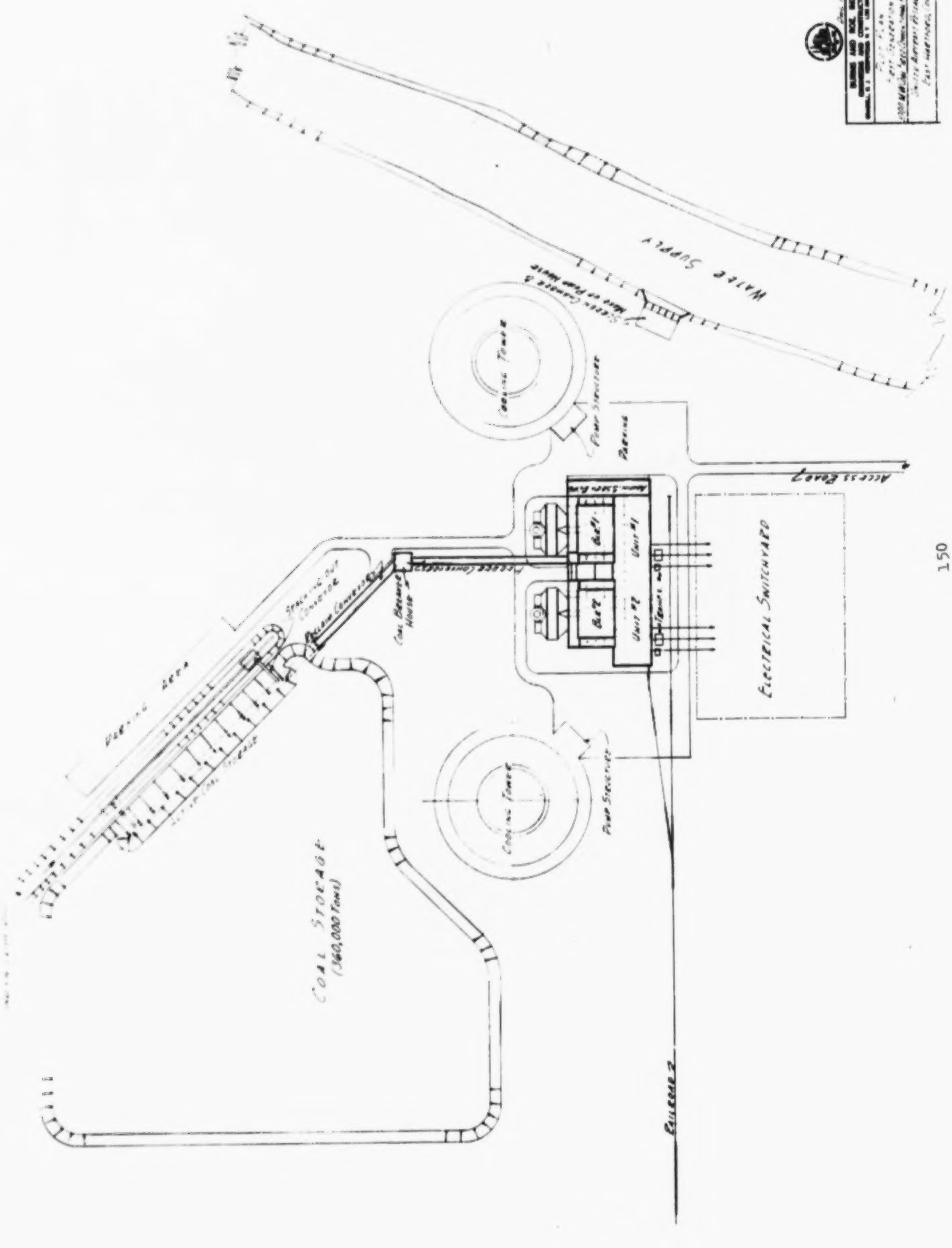

Coal Silos
Boiler No 1
Boiler No 2
Gas
Air
Air Preheater
Electrostatic Precipitator
Electrical Substation
Feed Water Heater No 2
Stairs
Control Room
Open
Fire Wall
Main Transformer
Aux Trans
Generator Rotor Removal Space



Deaerating Heater (Far Side)
Boiler
Coal Feeders to Silos
Air Preheater
Electrostatic Precipitator
Stack
F.D. Fan
Turbine Generator
Condenser
Electrical Transmission to Switchyard
Oper Floor El. 40'0"
Mezz Floor El. 20'0"
Main Transformer
Ground Floor El. 0'0"
Boiler Feed Pump (Turbine Drive)
Burns and Roe, Inc.
Engineers and Constructors
Cross Section
United Aircraft Research Lab
East Hartford, Conn.

FIG. 27

FIG. 28



WATER SUPPLY
SCREEN CHAMBER & MAKE-UP PUMP HOUSE
COOLING TOWER
PUMP STRUCTURE
PARKING
ACCESS ROAD
UNIT #1
UNIT #2
ELECTRICAL SWITCHYARD
COAL BREAKER HOUSE
STACKING OUT CONVEYOR
DUMPING AREA
COAL STORAGE (360,000 TONS)
RAILROAD
BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
UNITED AIRCRAFT RESEARCH
EAST HARTFORD, CONN.

PART TWO

ADVANCED-CYCLE POWER SYSTEM STUDIES

SUMMARY

An investigation was undertaken to estimate the technical performance and economic feasibility of advanced-cycle power systems which could be fueled with a desulfurized fuel. A review of current conventional steam power station technology was made to establish a basis for projections of future technology for this type station. The current and projected steam power system technology formed the basis against which all other advanced-cycle power systems were compared. Pertinent design specifications are presented for those systems judged to have the potential of generating power at busbar costs competitive with conventional steam power systems. Detailed economic characteristics of these elected advanced-cycle power systems are presented and estimated busbar power costs are compared.

The advanced-cycle power systems considered can be placed into two generic groups:

External Combustion	Internal Combustion
Conventional steam-electric	Open-cycle gas turbine
Topping and bottoming	Combined gas turbine-steam turbine
Closed-cycle gas turbine	

More specifically, the power systems investigated included: conventional steam power systems burning both treated and untreated coal and residual oil; potassium topping cycles and ammonia and fluorocarbon bottoming cycles; closed-cycle gas turbines with helium, carbon dioxide, and sulfur dioxide working fluids; simple-cycle gas turbines, and gas turbines with regeneration; and a variety of combined gas and steam (COGAS) systems. For the purposes of this study, a power system with a nominal capacity of 1000 Mw was selected as the basis for comparing performance and costs.

SYNOPSIS OF ADVANCED-CYCLE POWER SYSTEM STUDY RESULTS

The use of large industrial-type gas turbines utilizing aircraft-based technology in combination with a waste-heat recovery steam system appears to offer a means of generating electric power at busbar costs competitive with conventional untreated coal- or oil-fired steam systems even when burning a relatively high-cost desulfurized fuel. A first-generation combined gas and steam (COGAS) system would be competitive only at desulfurized fuel costs in the lower range of projected costs, but second- and third-generation COGAS systems would be competitive over a wide range of fuel costs.

Projected second- and third-generation simple-cycle gas turbine systems also appear to offer the promise of generating lowe-cost power than the conventional untreated coal- or oil-fired conventional steam power systems. While the projected busbar costs for base-load power from this type of system are not as attractive as those projected for the COGAS systems, a decrease in station load factor, high interest rates, and consideration of cooling water shortages would make the busbar cost of power from the gas turbine systems more attractive than that of the COGAS systems.

The above gas turbine and COGAS systems are based upon the attainment of turbine inlet temperatures of 2200 F and higher. Such temperatures can be attained if current aircraft technology in the fields of materials and blade cooling techniques is applied. In order to reach the 2800 F and 3100 F temperatures projected for second- and third-generation systems, the compressor bleed air used for cooling the turbine blades must be precooled to approximately 200 F in external heat exchangers before injection into the turbine section. This precooling would allow smaller bleed flows for a given temperature or a higher temperature for a given bleed flow. In either case, turbine performance would be enhanced.

The performance of conventional steam power systems is projected to remain nearly constant over the next two decades. Slight improvements in component efficiency can be expected, but increases in cycle operating conditions to give better performance cannot be economically justified.

Although several of the topping cycles and closed-cycle gas turbine systems offer significant improvements in performance over that projected for the conventional steam-power systems, the estimated capital costs for these systems, all of which required large amounts of high-nickel steels or refractory materials, were quite high. Thus the projected busbar costs were not as attractive as those for the COGAS and gas turbine systems.

DETAILED ADVANCED-CYCLE POWER SYSTEM STUDIES

Section I - Conventional Steam Power Systems

For many years, steam power systems have maintained a command position in the field of electric power generation. Consequently, this type of power system is the standard against which the feasibility of alternative methods of generating electric power must be compared. This section contains detailed design, performance, and economic analyses of projected future coal-fired and residual-oil fired steam power stations. Although these conventional fossil-fueled steam power stations would not necessarily meet future sulfur emission regulations, analysis of these stations is included to provide a basis of comparison for all other advanced power systems considered. The addition of stack gas cleanup equipment to these conventional fossil-fueled power stations presumably would enable them to meet near-term sulfur emission goals. However, estimation of the performance characteristics and costs of this additional stack gas equipment was beyond the scope of the present study, so that these considerations have not been included in the following analyses.

The economics of future steam power systems burning desulfurized fossil fuels have also been estimated. Only those desulfurized fossil fuels considered to become available during the three time periods spaned by the current study are included in the analyses. These desulfurized fossil fuels include hydrodesulfurized residual fuel oil for first-generation power stations, gasified residual oil for second- and third-generation power stations, and gasified coal for first-, second-, and third-generation power stations.

DESIGN CONSIDERATIONS

The design characteristics of future steam power stations are based on logical extension and extrapolation of technology used in present-day steam stations. In order to establish these characteristics, discussions were held with engineering, marketing, and product planning personnel from nearly all of the major power plant equipment manufacturers. In this manner, it was possible to assemble a complete picture of present and future steam station design technologies and economics exclusive of any major influence from a single information source. Future steam station designs were found to depend primarily on the output capacity for each single steam unit, the steam conditions utilized in these units, and environmental factors which must be considered during station design.

Unit Sizes

Unit size is primarily governed by the overall size of the electric utility system. A rough rule of thumb which has been applied to past capability additions is that the capacity of new units should not exceed 10% of the total utility system generating capacity. For a large electric utility system, such as TVA or American Electric Power, this 10% rule would dictate the selection of 1000-Mw units today. However, the basic guidelines of the present study call for analyses of 1000-Mw power generating stations installed in a typical, medium-size utility, for which 500-Mw units would be more representative of units installed during the 1970 decade. Consequently, two 500-Mw units were selected for first-generation power stations. It should be noted that the range of unit sizes installed during the 1970's will range from 100-Mw for some small municipal utilities to over 1300-Mw for the larger utilities.

Due to future utility growth, mergers, and cooperative intertie arrangements, 1000-Mw unit additions should be commonplace during the second- and third-generation time periods. Projections contained in Ref. 90 indicate that by 1980, many single-unit additions will be rated over 1000 Mw, with some unit ratings approaching 1400 Mw. The general feeling among industry representatives is that the economic incentive to build large units essentially disappears above approximately 1500 Mw. Consequently, single 1000-Mw units were selected for second-generation steam systems. During the 1990 decade, typical single-unit additions would exceed 1000 Mw, but because of the basic guidelines of the present study unit size was limited to 1000 Mw for third-generation systems.

The number of units and unit sizes selected for each generation are summarized in Table 41. The capacities indicated in this table are nominal capacities and do not represent the actual power-generating capacity of each unit. The gross power generating capability of the 500-Mw units was assumed to be 525 Mw. After substracting power required by the fuel-handling and other accessory equipment, net unit output would be approximately 500 to 515 Mw, depending on station design. Similarly, gross output for the nominal 1000-Mw units was assumed to be 1050 Mw. Net output from these units would vary from approximately 1010 to 1025 Mw, depending upon power station design.

Steam Conditions

Historically, improvements in steam station technology have permitted increases in steam temperature and pressure which resulted in increases in station efficiency and lower net fuel charges. However, as technology advanced more expensive equipment was required to contain the steam so that capital cost increased significantly. Until recently, the improvement in thermal efficiency and resulting reduced fuel charges always outweighed the higher capital costs for large-capacity units. At the present time, the highest practical steam temperature in new plants appears to have reached a plateau of approximately 1000 F with a single reheat to the same approximate temperature. A second reheat is not justified in present-day units because the capital costs would outweigh the marginal saving in fuel cost (Ref. 91). Presently, 2400 psig is the most common pressure in medium-sizes units, although several large units incorporating supercritical boilers operating at 3500 psig are now operating or are in the planning stages. Operating problems with the supercritical units, however, have not been completely overcome, so that the 2400 psig/100 F/1000 F steam cycle was assumed for use with the 500-Mw units selected for first-generation systems.

Equipment manufacturers were questioned specifically regarding the long-term trends in steam conditions. Surprisingly, there was unanimous agreement that there would be no increases in steam temperature beyond approximately 1000 F, nor increases in steam pressures beyond approximately 3500 psig. Previous experience with units rated at 1100 F to 1200 F (e.g., the Eddystone Station of Philadelphia Electric Co. and the Bergen Station of the Public Service Co. of New Jersey) was not encouraging, and these stations have been downrated to approximately 1000 F. Although these units were installed primarily to determine the reliability and performance obtainable at higher steam conditions, their continued operation at high temperature could not be justified economically. The basic problem with high-temperature operation arises because austenitic-type stainless steels must be used above 1000 F, whereas ferritic steels are adequate up to approximately 1000 F. Since austenitic steels are considerably more expensive than ferritic steels, high-temperature boilers could be very expensive, and the incremental cost of these boilers over boilers constructed from ferritic steels generally would not be offset by the incremental fuel saving. For example, several boiler manufacturers (Refs. 92 and 93), estimated that a boiler designed to generate 1100 F steam would cost approximately 8% to 10% more than a 1000 F boiler. Similarly, steam turbine manufacturers indicated that steam turbines designed for 1100 F and 1200 F would cost approximately 10 to 15% and 20 to 25%, respectively, more than steam turbines designed for 1000 F steam. Several equipment manufacturers (Refs. 91 and 94) conducted analytical tradeoff studies which indicated that these increased equipment costs would not be justified unless the fuel cost were to exceed approximately 45 to 50¢ per million Btu.

To substantiate the previous results reported by the equipment manufacturers, a simplified economic analysis was conducted to compare the economics of a 2400 psig/1000 F/1000 F cycle with a 4000 psig/1200 F/1200 F cycle. The materials and erection costs for the 1000 F boiler were estimated to be approximately $18/kw and $9/kw, respectively, for a total installed boiler cost of $27/kw. Of this total, approximately 35% or $9.50/kw would be for the expensive superheater and reheater components. The 1200 F boiler would require approximately 60% more superheater and reheater surface area than the 1000 F boiler. This additional surface would be constructed of austenitic steel which would cost approximately 3 times as much per unit of surface area as ferritic steels. The incremental cost for this additional surface area was estimated to be approximately $17/kw. The high-temperature system, however, would produce approximately 36.2% more output power than the low-temperature system, so that the adjusted boiler cost would be approximately $32.30/kw. The incremental cost over the 1000 F boiler, therefore, would be approximately $5.30/kw.

The cost of the steam turbine generator set for the 1200 F system was estimated to be approximately $26/kw compared to $20/kw for the 1000 F system, resulting in an incremental cost increase of $6/kw. Similarly, the high-pressure and high-temperature piping for the 1200 F system was estimated to cost approximately $6/kw more than for the 1000 F system. The total incremental cost for the high-temperature system, therefore, was estimated to be approximately $17.3/kw more than the cost of a 1000 F system.

In order for the 1200 F system to be economical, the savings in fuel cost must outweigh these additional capital costs. Consequently, a simplified performance analyses was conducted to determine the fuel cost reduction which would result from the higher efficiency operation of the 1200 F station. For the purposes of this analysis, the 1000 F station was considered to have a full-load efficiency of approximately 38.9%. The differential efficiency between the high- and low-temperature systems was estimated to be approximately 3.4 points in efficiency, resulting in a 42.3% efficiency for the 1200 F station. Figure 22 presents the results of a parametric study to evaluate justifiable capital cost increases for various fuel costs and efficiency increases using a 14% fixed charge rate and 70% load factor. From the results in this figure, it can be seen that a $17.3/kw capital cost differential and 3.4 point efficiency differential could be justified only at fuel charges greater than 53¢/million Btu, which is in close agreement with results obtained by the equipment manufacturers.

Consequently, 1000 F was selected as the peak cycle temperature for the second- and third-generation conventional steam stations burning untreated fuel. Since the 3500 psig supercritical cycles do show a slight increase in efficiency relative to cycles operating at 2400 psig, and since operational problems experienced with these supercritical pressures should be overcome by the 1980 time period, the 3500 psig pressure level was selected for conventional second- and third-generation steam stations.

The boiler manufacturers were also questioned regarding the possibility of future increases in steam temperature due to the prospective availability of clean, desulfurized fuels in the future. The boiler manufacturers indicated (see Ref. 93, for example) that fuel cleanliness has relatively little effect on maximum allowable steam temperature, although it does have a significant effect on boiler cost. For example, the materials required for a boiler fired with natural gas, which is an almost ideal fuel, cost approximately 43% less than the materials required for a coal-fired boiler, resulting in a net installed cost differential of approximately 40%. This cost differential results, primarily, because gaseous fuels do not produce any ash which could foul the boiler tubes or cause excessive material erosion. Therefore, gas-fueled boilers can be designed with smaller tube spacings, higher gas flow rates, higher heat transfer coefficients, and lower surface areas than coal-fired boilers. This potential cost differential notwithstanding, the boiler manufacturers indicated that the capital cost differential between gas-fired boilers designed for 1200 F and 1000 F, together with the higher cost steam turbine and steam piping, would not be compensated for by the fuel saving resulting from the increased operating efficiency. They pointed to the fact that the maximum economical temperature level for natural gas-fired stations is limited to approximately 1000 F. Consequently, steam conditions for future steam stations fired with desulfurized fuel were taken to be the same as the corresponding steam conditions for conventional coal- and residual-oil-fired stations.

Environmental Considerations

The effects of air and water pollution by power stations on the environment has caused consierable concern recently because of the substantial increases in power generating capacity projected for the next several decades. In order to minimize the ecological effect caused by using large amounts of river water for condenser cooling, power generating stations for all three generations were designed to incorporate hyperbolic, natural draft, evaporative cooling towers. The water requirements for these stations, therefore, would be reduced by approximately 95% because only makeup water for the cooling towers would be required.

Several equipment manufacturers indicated that even makeup water for evaporative cooling towers might not be available in some locations during the 1990 decade. In that case, nonevaporative cooling towers would be required to completely eliminate dependence on water makeup. Nonevaporative cooling towers utilize large areas of finned tube heat exchanger surface to reject the condenser water heat to the ambient air. Since these towers require large heat transfer surface areas, they could cost as much as $30 to $40/kw. Because of the relatively poor heat transfer between the cooling tower water and the ambient air, larger temperature differentials would generally be desired to increase the heat transfer rate. As a result of these larger temperature differentials, the temperature in the steam condenser would increase above the levels normally employed. The turbine back pressure would also increase, to as much as 6 to 10 in. of mercury, thereby reducing output and overall power system efficiency. While nonevaporative towers might be required at some restrictive sites, these restrictive sites were not considered to be typical. Accordingly, evaporative cooling towers enabling operation at 2 in. mercury turbine backpressure were assumed for all generations.

As previously indicated, sulfur removal from stack gases was not considered for the steam power systems analyzed herein, so that only the particulate matter in the boiler effluent was considered to be a problem. As a result, a small-diameter mechanical cyclone dust separator followed by a 600 ft stack would be adequate for residual-oil-fired stations. Coal-fired stations, however, would require the utilization of electrostatic precipitators and taller, 750-ft stacks to disperse the flue gases. With this type of air pollution control, coal- and residual-oil-fired stations would not meet the projected sulfur emission goals. Nevertheless, these polluting power stations were used as a basis of comparison of future advanced-cycle, nonpolluting power stations.

CHARACTERISTICS OF FUTURE STEAM POWER STATIONS

The projected design, performance, and economic characteristics of conventional coal- and residual-oil-fired steam power stations are presented in the following paragraphs for the three time periods of interest. Corresponding projections are also presented for steam power systems utilizing clean, desulfurized fuel. These projections are based on generation of 1000-Mw nominal output.

Power Station Designs

The basic design characteristics for the first-generation conventional coal- and oil-fired steam power stations are summarized in Table 42. The oil-fired station was assumed to be located in the northeastern US near a navigable river capable of accommodating small tankers for fuel delivery. The station would be capable of burning No. 6 Venezuelan residual oil, the specifications for which are summarized in Appendix B. As previously indicated, two 500-Mw units with 2400 psig/1000 F/1000 F steam conditions would be required. The steam turbines would be 3600 rpm, tandem-compound, 4-flow machines with 30-in. last-stage blades. There would be provisions for 7 stages of extraction for feedwater heating. In addition, steam would be extracted from the low-pressure crossover pipe at approximately 185 psia to supply steam to the two boiler feed pump turbine drives. Boilers and boiler auxiliaries would be completely enclosed in a building. Each boiler would be pressurized by two half-capacity enforced-draft fans, and would be equipped with steam atomized oil burners, an air preheater, soot blowers, an economizer, and a 100%-capacity, condensate polishing demineralizer to prevent carry-over of dissolved solids. The steam would be condensed in a single-pressure, two-pass, twin-shell condenser wherein the tubes would be perpendicular to the turbine center line. Steam from the boiler feed pump auxiliary turbine drives would also be condensed in the main condensers.

A detailed layout of the operating floor of a typical two-unit, oil-fired station is presented in Fig. 23. A cross-section of the boiler and power plant buildings is presented in Fig. 24, and a plot plan showing the relative location of the power house, fuel storage facilities, and electrical switchyard is presented in Fig. 25.

Since desirable construction sites in the northeastern US are limited, two types of construction sites were considered. The first type of construction site would require maximum expenditure for site preparation, excavation, and pilings, whereas the second site would require only a minimum allowance for site preparation and construction. Both sites would require considerable architectural treatment to yield an attractive plant.

Representative design characteristics for the first-generation coal-fired steam power station are also summarized in Table 42. This station was considered to be located at a typical mine-mouth site such as might be encountered in southern Illinois. This station would be designed to burn a southern Illinois Seam No. 6 coal, the specifications of which are presented in Appendix B. The coal would be conveyed directly from the mine to the power plant where it would be stock piled in storage piles or fed through pulverizing mills directly to the boiler. The conveyer system would be sized to accommodate approximately 400 tons of coal per hour, the approximate full-load requirement of a 1000-Mw station. With the exception of the slightly taller stack required for coal-fired stations, other station equipment would be identical to that previously described for the oil-fired station. Detailed operating floor plans, cross-sections, and plot plans for the first-generation coal-fired station are presented in Figs. 26, 27, and 28, respectively.

Design characteristics for second- and third-generation coal- and oil-fired steam stations would be similar to those previously described for first-generation stations with the exception that a single 1000-Mw nominal 3500 psig/1000 F/1000 F steam unit would be required. Future steam stations designed to burn desulfurized liquid fuel such as hydrotreated oil would essentially be identical to stations designed for residual fuel oil. Future steam stations designed to burn gasified coal or gasified residual oil would essentially be the same as stations designed for natural gas. For these stations, the only changes required would be in the boiler, fuel handling facilities, and standby clean oil storage facilities. Hydrotreated residual oil and Fuel Gas II were selected for first-generation oil- and coal-fired stations, respectively, Fuel Gas IV and Fuel Gas III were selected for second- and third-generation oil- and coal-fired stations, respectively. The characteristics of these prospective low-Btu fuel gases are presented in Appendix F along with temperature rise data and the properties of the products of combustion.

Performance

Although the steam temperature is projected to remain essentially constant throughout the entire 30-year time period under investigation, slight performance improvements are anticipated because of the increase in pressure level between first- and second-generation systems and also because of slight improvements in steam turbine and boiler efficiencies during these time periods. Projected performance characteristics for future steam power stations are presented in Table 43. As previously noted, the total gross output for each power system would be approximately 1050-Mw. Differences in power station auxiliary power requirements account for the slight differences in the net outputs among the coal-, oil-, and gas-fueled power systems. As noted in the table, oil-fired power systems would have slightly higher efficiencies than coal-fired systems, primarily because of the lower auxiliary power requirements. Power systems fueled with gasified coal or gasified oil would have still higher efficiency, primarily because of the lower stack gas temperatures which would be obtainable due to the use of a desulfurized fuel.

Since conventional coal-, and residual-oil-fired power systems would require no fuel conversion process, the net power station performance would be the same as that shown for the power system. Those steam power systems requiring the use of a desulfurized fuel must be used in conjunction with a fuel treatment process. These fuel treatment processes might consume some of the electricity generated by the power system. In addition, thermal losses during fuel conversion would reduce the net efficiency of the integrated power station. The net estimated power station outputs and efficiencies are also tabulated in Table 43. The efficiencies included in this table are all average annual efficiencies based on a 70% load factor. As such, they are approximately 5% less than the design efficiency levels.

Economics

Detailed capital and operating cost estimates are presented in the following paragraphs for conventional coal- and residual oil-fired steam power stations. Subsequent estimates for future steam power stations are based on these detailed first-generation cost estimates with appropriate corrections applied to account for differences in unit size, boiler design, and operating efficiency.

Capital Costs

Detailed capital cost breakdowns for first-generation conventional coal- and residual-oil-fired steam power stations are presented in Tables 44 through 50. These detailed cost breakdowns are presented according to standard Federal Power Commission (FPC) account categories which include land and land rights, structures and improvements, boiler plant equipment, turbogenerator units, accessory electric equipment, miscellaneous power plant equipment, and miscellaneous station equipment. Summaries of these first-generation stations costs are presented in Table 50. Also included in the cost summary are other expenses associated with start-up, testing, temporary facilities (for heat, light, power, water, fire protection, etc.) temporary buildings (warehouses, office, change house, toilet, first aid center, and garage) removal of temporary facilities, clean-up, security guards, etc. Indirect construction expenses for engineering, design, construction supervision, contingency, escalation, and interest during construction are also shown in this summary table. Capital cost summaries for second- and third-generation conventional coal- and residual-oil-fired steam stations are presented in Table 51.

Capital costs for steam power systems designed to burn a desulfurized liquid fuel, such as hydrotreated oil, were assumed to be the same as those presented for residual oil-fired stations. Steam power systems designed to burn a gaseous fuel derived by gasification of coal or residual oil would have essentially the same cost as a natural gas-fired station with the exception that a standby fuel would have to be provided in the event that the fuel conversion processes failed or was shut down. This standby fuel was assumed to be a low-sulfur oil. The incremental costs which would have to be assigned to the costs of gas plants for standby oil burning capacity are summarized in Table 52. This cost estimate includes the incremental cost for the boiler because the boiler would have to be designed to burn both gaseous and liquid fuels. Capital cost summaries for first-generation gas-fueled steam stations, including the incremental cost for standby oil, are summarized in Table 53 for power systems constructed at sites requiring maximum and minimum construction expenses. Capital cost estimates for second- and third-generation gas-fueled steam power systems are summarized in Table 54. Three cost estimates are presented which were obtained by applying appropriate corrections to the previous estimates presented in Table 51 for coal- and residual-oil-fired power stations. The resulting cost estimates were averaged to provide a final estimate which, for all practical purposes, is independent of site location. The required incremental cost for standby storage of oil was added to obtain the total estimated cost for the gas-fueled steam power systems shown in Table 54.

Annual Owning and Operating Costs

Summaries of capital costs and annual owning and operating costs for first-, second-, and third-generation steam power stations are presented in Tables 55, 56, and 57, respectively. Where appropriate in these tables, capital cost estimates for the fuel processing system have been added to the capital cost estimates for the power systems to arrive at the total estimated station cost. The resulting net station costs, per unit of net station output power, are also shown in these tables. The annual capital charges are based on 14% capital charges and 70% load factor. Raw fuel costs were presumed to be 20¢/million Btu for coal and 30¢/million Btu for residual oil. The final busbar power costs for all steam power stations are shown in these three tables.

The cost of power from conventional coal-fired steam power stations is estimated to decline from approximately 6.227 mills/kwhr to approximately 5.714 mills/kwhr during the time span under consideration, provided no stack gas treating facilities would be required for sulfur removal. Corresponding power costs from residual-oil-fired stations would decline from an estimated 6.969 mills/kwhr to 6.432 mills/kwhr for those sites requiring maximum construction expenses. Those sites requiring minimum construction expenses would yield power costing approximately 0.15 mills/kwhr less for each generation. The cost penalty for obtaining clean air from steam power systems by utilizing gasification of coal is estimated to be approximately 2.4 mills/kwhr for first-generation systems. This penalty would decline to approximately 1.0 mill/kwhr for third-generation systems. Similarly, the cost penalty associated with using hydrotreated oil for pollution control in first-generation systems is estimated to be approximately 1.1 mills/kwhr. The cost for hydrotreated oil is based on the use of a small 34,000 bbl/day (nominal) hydrotreating facility located adjacent to the power system, with credits taken for middle distillate. The cost penalty for second- and third-generation oil-fired stations utilizing gasification of oil would remain at approximately 1.0 mill/kwhr.

Section II - Open-Cycle Gas Turbine Power Systems

This section contains the results of a study undertaken to identify the design, performance, and economic characteristics of large gas turbine incorporating advanced-design features derived from aircraft gas turbine engines for base-load electric power generation. Presently, the utilization of gas turbine engines for stationary electric power generation is limited to peaking power applications because of the gas turbine's relatively low thermal efficiency in comparison to fossil-fuel steam-electric power plants. However, recent engineering advances achieved during extensive research and development efforts on military and commercial aircraft applications has provided the basis for projecting substantially improved large-capacity base-load gas turbine power systems with significantly higher thermal efficiencies than those attainable with present systems. Based upon these projections of the rate of progress of gas-turbine technology for the next two decades, parametric performance analyses were performed for the simple-cycle turbine and also for regenerative cycles. Pertinent size and detailed cost characteristics of 200- to 500-Mw base-load gas turbines for use in 1000-Mw power stations are presented. In addition, estimates of the cost of producing electric power with advanced open-cycle, base-load gas turbines were made in terms of the net cost of fuel delivered to the power system.

The results of the analyses of open-cycle gas turbine power systems indicate that the introduction of aircraft gas turbine technology into base-load gas turbines will allow significant improvement in performance and that advanced-cycle gas turbine power systems should receive consideration as a source of base-load electric power.

DESCRIPTION OF BASE-LOAD GAS TURBINE DESIGNS

A review of industrial gas turbine designs built over the last 20 years shows that they tended to oscillate between complex reheat, intercooled, and/or regenerative cycles, offering relatively high thermal efficiencies, and simple cycles offering reliability but relatively low thermal efficiencies. The continued re-emergence of the simple-cycle gas turbine indicates that, despite the effort expended, the industry has failed to provide a completely satisfactory high-efficiency prime mover. Much of the trouble encountered in the development of the more complicated cycles for relatively large gas turbines has been due to miscalculation of pressure losses, particularly in ducts and elbows, and to the high additional expense of heat exchange equipment (Ref. 95).

Recent advances in turbomachinery, including materials technology, aerodynamic design, heat transfer, and fabrication techniques, achieved during extensive research and development efforts on military and commercial aircraft engines has provided the basis for substantially improved large-capacity, base-load, simple-cycle gas turbine power plants with significantly higher thermal efficiencies than are attainable with present systems. Pertinent aspects of these technological advances, including their effect on industrial gas turbine designs, and projections of further design advances anticipated during the next two decades are discussed in the following paragraphs.

Projections of Base-Load Gas Turbine Materials Technology

Existing differences between design philosophies of industrial gas turbine power plants and aircraft gas turbine propulsion systems will diminish as a result of continuing efforts to improve thermal efficiency. Significant improvements in specific horsepower (hp/lb-sec of engine airflow) and thermal efficiency for simple-cycle gas turbines can be achieved only by increasing turbine inlet temperature and compressor pressure ratio, since turbomachinery component efficiencies have reached a relatively high level due to over 30 years of extensive research and development on compressors and turbines for aircraft propulsion.

Progression of turbine inlet gas temperature with time for base-load gas turbine designs is shown in Fig. 29. The lower curve in this figure shows that improvements in turbine materials have historically afforded a 20 F/yr increase in turbine inlet gas temperature. The upper shaded curve reflects the technology expected to be available during the prescribed time period, based on improved materials and turbine cooling techniques. Superimposed on Fig. 29 are also base-load turbine inlet gas temperatures for existing power plants and for power plants currently under development and expected to be commercially available during the next decade; also shown are projections by industrial gas turbine manufacturers of anticipated base-load design technology for the next two decades (Refs. 96 and 97). Turbine blade cooling methods that will permit operation at the projected gas temperatures are shown along the ordinate on Fig. 29. Technological advances in aircraft compressor design hve resulted in substantial improvements in compressor pressure ratio as depicted in Fig. 30. The technology is becoming available that will allow the introduction of machines with pressure ratios exceeding 30, a prerequisite for high-efficiency base-load gas turbine cycles for use in second-generation power plants.

Turbine Materials

Part of the increase in turbine inlet gas temperature has been permitted by improved turbine materials. An indication of the rate of improvement in the elevated temperature capabilities of turbine blade and vane materials for aircraft gas turbine propulsion systems is shown in Fig. 31. The gains have been most significant for the nickel-base alloys, with an improvement of more than 300 F in temperature capability over the past decade (Ref. 98). Presently nickel-base cast blade alloys, such as B-1900 and International Nickel Company's IN-100, have, in some propulsion systems, replaced forged Udimet 700 (U-700) for high-temperature applications, at the same time decreasing thermal-fatigue failures by a factor of five.

It should be noted that these materials were developed for relatively short-time (1000-hr) high-strength applications. Upon long-time exposure (approaching 100,000 hr) at relatively high metal temperatures, as required in base-load electric power generation, these high-temperature nickel-base alloys have a tendency to form a new "sigma" phase. This new constituent in the alloy is developed during service and subsequently causes the alloy to become brittle and to experience a loss in rupture-strength capabilities (Ref. 99). It has been determined that, for many of these nickel-alloy materials, the occurrence of this metallurgical instability during long-time exposure can be avoided through modifications to their heat-treatment cycles (Refs. 100 and 101). Presently, wrought Udimet 500, heat-treated for long-life, high-strength capabilities, is used in many high-performance industrial designs, and cast Udimet 500 and Udimet 700 hollow blade designs are being exploited for utilization in more advanced designs.

Second-generation turbine blade materials for industrial designs will include the high-temperature nickel-base alloys currently under development for advanced aircraft gas turbine propulsion systems; these include a modified B-1900 A alloy and unidirectionally solidified eutectic alloys, such as Ni_3Al-Ni_3Cb (Ref. 102). Although it is impossible to predict with accuracy the materials and associated creep strength properties that will be available for the third-generation designs, it should be reasonable to expect a continuation of at least a 20 F/yr improvement in material temperature. Chromium-base alloys, for instance, with potential firing temperatures to 200 F (Ref. 99) are currently being investigated. Recent developments in the ability to coat columbium-base alloys offer a possibility for the use of these alloys, currently being investigated for second-generation aircraft gas turbine propulsion systems, in third-generation industrial power plants.

The predicted long-time creep-strength characteristics (1% creep in 100,000 hr) for three generations of advanced turbine-blade materials are shown in Fig. 32. The first- and second-generation alloy creep strength properties were estimated by applying Larson-Miller type extrapolations to the short-time properties of the respective materials. The third-generation material estimated creep-strength properties reflect a 20 F/yr rate of improvement in material properties. As previously mentioned, modified heat treatment cycles would have to be developed for these short-time, high-strength alloys to provide them with the 100,000 hr creep-strength properties depicted in Fig. 32. Also shown in Fig. 32 for comparative purposes are estimated creep strength properties for two cobalt-base alloy (SM-302 and AMS 5382) that have frequently been used in aircraft gas turbine designs.

Creep strength improvements for vane alloys have not been as great as they have for blade alloys (see Fig. 31), and WI-52, a cast cobalt-base alloy, has been a primary vane alloy since 1958. Nickel-base alloys have been used for vanes in some power plants to take advantage of their relatively high strength and improved thermal-fatigue resistance characteristics, but generally they have shown relatively poor oxidation-corrosion resistance by comparison with cobalt-base alloys, and they require relatively difficult weld repair procedures.

Coatings have been developed which provide adequate corrosion resistance for the blades and vanes. The two most successful coatings, both aluminum base, have been Type UC (developed by Chromalloy Corporation) for WI-52 and "Jo-Coat" for nickel-base alloys. Although blade and vane coating lives in aircraft propulsion system designs are measured in terms of a few thousand hours, considerably longer coating lives could be attained in industrial designs through the application of coatings with thicknesses greater than those used in the aircraft versions (Ref. 103).

Turbine disks in current industrial power plants that operate at gas temperatures of 1600 to 1700 F are maintained at metal temperatures on the order of 600 to 750 F (Ref. 104) by the utilization of cooling air extracted from the compressor. As a result of these moderate disk temperatures fairly inexpensive materials such as austenitic iron-base alloys, used for years in aircraft designs, have also been used in the stationary designs. Industrial turbine manufacturers utilize CR-MO-V (a low-alloy steel) alloy in their intermediate- and large-size (greater than 36 in. dia) turbine disks; this material is the same as that used in their large steam turbine rotors (Ref. 99). As base-load turbine inlet gas temperature is increased to levels on the order of 2000 F and above, disk materials similar to those used in aircraft propulsion systems and capable of withstanding metal temperatures as high as 1200 to 1400 F will have to be utilized. Typical materials are IN-100 for operation up to 1400 F and Incoloy 901 (austenitic iron-nickel alloy) for operation at temperatures from 900 to 1300 F.

Compressor Materials

In aircraft gas turbine propulsion systems where light weight is a primary design criterion, titanium alloy, such as Ti-6Al-4V (AMS 4928), is used extensively for compressor vanes, blades, and disks subjected to temperatures not exceeding 750 F. Incoloy 901 is commonly used in the high-pressure stages of relatively high-pressure-ratio compressor designs where rim temperatures might exceed 750 F. Industrial gas turbines, which normally experience compressor rim temperatures below 800 F, utilize a martensitic corrosion resistant steel (AISI Type 410) for blade and vane material and a low alloy steel such as AISI 4340 for disks. It would appear that in the next decade, large-capacity base-load designs will continue to use AISI 4340 and Incoloy 901 alloys for disk materials in the low-pressure-ratio and high-pressure-ratio stages, respectively (as determined by the aforementioned temperature limits). In order that desirable blade stiffness can be maintained in the initial stages of the low-pressure compressor of these large units, without having to reduce blade aspect ratio below approximately 3.0 and/or revert to the use blade stiffeners (thus compromising efficiency), it might be desirable to substitute AMS 4928 for AISI Type 410 as blade and vane material in appropriate stages; Incoloy 901 alloy would be used in the relatively high (greater than about 750 F) temperature region stages.

Major changes in compressor materials for the third-generation base-load designs could be stimulated by efforts to fabricate lightweight and stiff blades, features that permit blade designs with aspect ratios greater than 3.0 (thus reducing mass and, hence, cost of both blade and supporting disk) without having to compromise performance through the use of stiffening rods. Among the most important materials offering these features are the fiber-reinforced composite materials. The two most promising fibers at the present time appear to be silicon carbide-coated boron fiber and carbon fiber. Carbon/epoxy (for temperatures less than 240 F), carbon/polymide (for temperatures of 250 to 600 F) and Borsic®/Ti (for temperatures greater than 600 F) composites are currently being developed for compressor components in aircraft engines and should be economically feasible in industrial units designed for third-generation use.

Casing Materials

The inlet and compressor casings for typical industrial designs are fabricated from cast iron, whereas the turbine shell is usually fabricated from nodular iron (Ref. 105). These types of materials should also suffice in the high-temperature, high-pressure-ratio advanced base-load gas turbine designs, provided external cooling (more likely with water) of the turbine casing is employed.

Combustor Materials

A typical material for combustor cans in industrial designs has been AISI Type 310, an oxidation-resistant austenitic stainless steel. However, in the advanced first- and second-generation high-temperature (above 2000 F turbine inlet temperature) base-load designs, materials such as Hastelloy X (allowable metal temperature of 1600 F) or perhaps coated TD nickel (permitting a 200 F increase in allowable metal temperature) will be used. Further improvements in combustor materials for third-generation designs could include the use of coated refractory metals such as corrosion-resistant columbium alloys with relatively large hafnium contents.

Advances in Turbine Cooling Techniques

Only first-stage vanes and disks of current advanced-design industrial gas turbines, operating at 1600 to 1700 F turbine inlet gas temperatures, are cooled. For long-life, base-load operation at turbine inlet temperatures of 1800 F and above, succeeding stages of turbine blades and vanes will also require cooling. Turbine cooling can be accomplished with coolants such as air, water, or liquid metals, but because of the complex cooling system designs and mechanical problems associated with liquid systems air has been used, exclusively, as a coolant in all aircraft propulsion systems and for most stationary applications.

Turbine cooling systems have progressed from simple convective cooling configurations incorporating cast round radial passages for vanes and single cavities for blades (Fig. 33) to advanced convective heat transfer designs utilizing impingement cooling of the inside surface of the leading edge (Fig. 34). The cross-section of a typical advanced impingement-convection-cooled blade for an aircraft engine is shown schematically in Fig. 35. In this configuration cooling air enters a central cavity within the airfoil. Since the blade tip is capped, all of the cooling air flows out of the central cavity in a forward direction; the air is forced through a series of orifices and impinges as high-velocity jets upon the inside surface of the leading edge. Upon cooling the leading-edge region, the cooling air enters a series of small finned passages which extend chordwise in the mid-chord region of the airfoil and cools, at high velocity, the mid-chord region of the airfoil by forced convection. The cooling air is finally ejected from the extreme trailing edge into the gas stream.

Advanced impingement-convection cooling techniques should afford base-load operation at turbine inlet temperatures as high as approximately 2400 F. Film cooling (hollow blades with radial slots through which is injected a layer of coolant forming an insulating air blanket for the surface to be protected) and transpiration cooling (porous material which in practice may consist of small drilled holes along the airfoil surface through which coolant is bled) must be relied upon to achieve operating temperatures approaching 3000 F. These schemes (see Fig. 34) are currently under investigation for aircraft propulsion systems and should be economically feasible for third-generation industrial power plants.

For stationary power plants, water-cooled heat exchangers could be used to reduce the temperature of compressor discharge cooling air from the 800 to 1200 F temperature level to a fairly low level (125 to 250 F). With this technique, it should be possible to increase the turbine inlet gas temperatures to higher levels and/or take advantage of relatively simple blade configurations. Such a design, shown schematically in Fig.35,could possibly be cast in one piece, except for the tip cap that would be required.

Projections of Base-Load Gas Turbine Component Performance Characteristics

Although the most significant advances in gas turbine propulsion systems have been in improved materials and turbine cooling techniques, further improvements have also been made in component efficiencies, turbomachinery stage loadings, and combustion processes.

Compressor Component Performance

Technological advances in aircraft compressor design have resulted in significant improvements in compressor efficiency, from an average of about 86% (polytropic) in 1950 to approximately 90% in current designs; further improvements to peak values of 91 to 93% appear feasible (Ref. 101). Compressors designed in accordance with aircraft technology exhibit an average stage pressure ratio of 1.29, whereas their industrial counterparts, designed with moderate blade loadings and rotative speed develop an average stage pressure ratio of 1.12 (Ref. 101). These stage pressure ratios are achieved with airfoil diffusion factors (D_f) of approximately 0.45 for aircraft compressor designs and 0.40 for the industrial versions. High levels of D_f are desirable if end-wall losses and surge margin are not affected, since this would reduce the number of compressor stages, assuming the other parameters were held constant. Since the present levels of compressor efficiency are relatively high, current aircraft compressor research efforts are directed primarily at increasing stage pressure ratio without sacrificing efficiency.

Principal factors which influence stage pressure ratio are rotor tip-speed and work coefficient; an increase in either or both parameters is desirable. Corrected rotor tip speeds as high as 1100 ft/sec, without compromising component efficiency, are attainable with current advanced lightweight compressor designs; tip speeds as high as 1200 ft/sec will be achieved in second- and third-generation designs. However, future large base-load power plants will likely utilize derated compressor stage designs to avoid prohibitive development costs. Therefore, in first-generation designs, attempts will not be made to exceed corrected tip speeds of 1000 ft/sec; this value will probably increase to 1100 ft/sec for second-generation designs and to 1200 ft/sec for third-generation designs. In addition, efforts will not be made to design for the high stage loadings achieved in lightweight aircraft-type compressors, since again compactness and light weight are not as important in the design of base-load systems as high efficiency and low development costs.

Compressors in aircraft gas turbine propulsion systems are designed for specific flows (lb/sec/flow area) which may vary from approximately 33 to 40 lb/sec/ft^2. Increases in this parameter result in more compact designs (smaller cross-sectional area); however, for a given wheel speed and work coefficient, higher specific flows and alternatively higher axial velocities mean high flow coefficients (C_x/U_m) and correspondingly lower compressor efficiency. There does not appear to be any advantage, particularly for industrial applications, in increasing specific flows above approximately 35 to 38 lb/sec/ft^2.

Combustor Component Performance

Combustion chamber efficiencies of 100%, currently obtained with aircraft gas turbine propulsion systems, can also be obtained in advanced industrial designs, particularly in those burning clean, gaseous fuels. As the turbine inlet gas temperature of advanced industrial gas turbines is raised to higher levels, existing differences between the design philosophies of industrial- and aircraft-type combustors will diminish considerably. For instance, multiple-can combustors currently used in industrial designs will have to be replaced by annular configurations in second- and third-generation industrial units operating at turbine inlet gas temperatures above approximately 2000 F in order to avoid excessive liner cooling flow penalties. Reference velocities, defined as the theoretical velocity for flow of combustor-inlet air through an area equal to the maximum cross-section of the combustor casing, of 80 to 100 ft/sec with corresponding combustor pressure losses of 3 to 8% are common in present aircraft designs; values of 50 to 80 ft/sec with correspondingly lower pressure losses are found in multiple-can industrial designs (Ref. 104). Increases in turbine inlet gas temperature of industrial designs to values of 2000 F and above can force reference velocities for these designs to levels of 125 to 150 ft/sec (with a corresponding increase in pressure loss) in order to avoid unstable radial temperature profiles in the first-stage turbine vanes. Conversely, however, the development of new combustor liner materials, such as the aforementioned coated refractory metals, might permit reference velocities to remain at present levels without the imposition of excessive cooling flow penalties. The length-to-height ratio of combustors has varied from 2.7 to 5, but generally lies between 3 and 3.5. Burner length is determined by two factors: length required for combustion and length required for dilution or mixing of unburned gases and combustion gases.

It appears from preliminary studies conducted at UARL that the formation of nitrogen oxides does not occur until after all the heat has been released in the combustion process. Consequently, the extremely short products of combustion lapse times associated with annular combustors, relative to multiple-can combustors, will be beneficial in the inhibiting formation of this pollutant.

Turbine Component Performance

Technological advances in aircraft turbine design have paralleled those for the compressor, with turbine isentropic efficiency increasing from about 85% in early 1950 designs to over 90% in current advanced designs; efficiencies of 92 to 93% are feasible in the near future (Refs. 95 and 101). The stage work for aircraft-type turbines may vary from as little as 20 Btu/lb in a low-pressure-ratio turbine to a maximum of about 155 Btu/lb in present advanced-design, high-pressure turbines.

Differences in design philosophies between aircraft- and industrial-type turbines will, as in the case of combustor design trends, decrease with predicted increases in turbine inlet gas temperature. For instance, future industrial turbine designs will incorporate a higher degree of impulse staging than exists in most present designs to achieve a relatively large gas temperature drop in the nozzle, thus avoiding high gas temperatures in the rotor and correspondingly high cooling flow requirements (Ref. 100).

High stage work in itself does not cause aerodynamic problems, but when associated with high efficiency, it requires high wheel speeds which may cause structural problems. Turbine tip-speeds in current advanced-design aircraft propulsion systems may vary from 900 to 1000 ft/sec in the last stage of the low-pressure turbine to as high as 1500 to 1600 ft/sec in the high-pressure stage. These tip speeds can be achieved in future industrial designs that utilize advanced turbine materials described in the preceding sections of this report.

Base-Load Gas Turbine Fabrication Techniques

The following paragraphs present a brief review of some of the pertinent factors that should be considered in establishing the design configuration of high-temperature, high-pressure, gas turbines for base-load electric power generation applications. Since light weight and compactness are secondary to high efficiency, low cost, and long life as design criteria in stationary applications, compressor rotor construction for advanced-design base-load power plants will be similar to that of conventional long-life, relatively inexpensive industrial designs; such a design is shown schematically in Fig. 36. The rotor for each compressor stage consists of a solid disk. Through-bolts connect the rotor disks to the forward and aft subshafts to form a disk-drum rotor assembly. The bolt circle is placed at the maximum diameter possible within the confines of the rotor design to ensure a strong, stiff rotor assembly. The compressor disk center rabbet holds the disks radially in alignment. This disk-drum type configuration eliminates the need for labyrinth seals at the blade root section between compressor stages, since a close-fitting hub is formed by the adjacent rims of the stacked disk assembly.

Compressor blade aspect ratios (defined as blade height divided by mean chord) of 3.0 or less are desirable in the initial stages to avoid the use of stiffening rods (which contribute to efficiency loss) that would otherwise be needed to prevent blade flutter. An aspect ratio of at least 1.0 is desirable in the last stages to minimize clearance losses. Compressor blades are usually machined to the required shape; therefore, a material with a machineability index greater than 16 to 20% (aluminum machineability equals 100%) is desirable in order to facilitate the use of conventional machining techniques (Ref. 105). The vanes in the low-pressure end of the compressor can be inexpensively fabricated from rolled and welded sections as opposed to the integrally cast vane sections used in aircraft engines. The rolled sections can be made in any size range and, because of the positive attachments to the inner and outer vane rings through welded joints, these sections can be stronger than equivalent ones employed in aircraft engines (Ref. 105). In the high-pressure stages where relatively high-temperature considerations require the use of Incoloy 901 alloy, cast vane sections will be required.

As previously mentioned, gas turbines designed to operate with turbine inlet temperatures exceeding 2000 F will have to utilize annular combustor configurations. Conventional louvered combustor liners will be permissible for turbine inlet gas temperatures approaching 2400 F; however, at temperatures of 2400 F and above advanced design liners with high-heat-transfer surfaces, such as the Thermal Skin® configurations developed by Pratt & Whitney Aircraft Division for aircraft engine designs will be required. The experience gained from related annular combustor designs in aircraft propulsion systems indicates that the use of these combustor configurations will permit liner lives comparable to those obtained with conventional multiple-can configurations. The annular combustor design is also well-suited to rapid maintenance, since the complete engine does not have to be disassembled to change the liner.

The turbine design for advanced base-load gas turbine power plants will be similar to its aircraft counterpart, since this component must be capable of operating at high temperatures for long periods of time in order to achieve the desired thermal efficiency levels. These base-load turbines will not experience the thermal stresses that are prevalent with the aircraft designs; consequently, they could be more adaptable to the utilization of supplementary cooling techniques, such as the use of liquid cooling or alternatively the use of precooled compressor bleed air, thus reducing turbine cooling flow requirements.

Journal bearings will be applicable in advanced power plant designs provided compressor tip speeds do not exceed approximately 900 ft/sec. However, at tip speeds on the order of 1000 ft/sec and above, oil-dampted roller bearings, which also eliminate critical speed problems frequently occurring in relatively long length engine designs, are recommended.

ANALYSIS OF ADVANCED BASE-LOAD GAS TURBINES

The anticipated advances in design technology and materials improvement projections described in the preceding section formed the basis for parametric performance and cost studies of large 200- to 500-Mw base-load gas turbine power plant designs. Both simple-cycle power plant designs incorporating either single-spool or twin-spool turbomachinery configurations and regenerative-cycle power plant designs with single-spool configurations were investigated (see Fig. 37).

The principal fuel assumed in thse studies as a low-heat-content (HHV = 162 Btu/ft^3) gasified coal derivative (Ref. 106) referred to as Fuel Gas I (see Appendix F). The coal-gasification process, fuel composition and thermodynamic properties, and the fuel processing cost characteristics have been described in Part One. In order to assess comparative performances of power plants burning this fuel and a high-heat-content gaseous fuel, a brief parametric performance study of power plants burning methane (HHV = 1000 Btu/ft^3) was also conducted.

Parametric Performance Analysis

The primary power plant independent variables (namely, turbine inlet gas temperature, compressor pressure ratio, and where applicable regenerator effectiveness) and the various turbine cooling techniques considered are presented in Table 58. The power plant design parameters and constraints, reflecting the design technology and materials improvement projected to be available during the early 1970's (first generation), early 1980's (second generation), and early

1990's (third generation) listed in Table 59 were used to determine the performance and power plant dimensions in these parametric studies. Turbine blade cooling configurations commensurate with anticipated advances in the state of the art for the projected time period (see Table 58) were assumed, and the associated penalty to the gas turbine power system performance was appropriately assessed.

A high-speed digital computer program developed in Ref. 107 was used to facilitate these parametric studies, to provide a realistic assessment of turbine cooling flow penalty effects on power plant thermal efficiency and specific output (hp/lb/sec), and to determine the pertinent sizes and dimensions of hypothetical power plant systems for use in power plant cost evaluations studies.

A flow chart of the computer program is shown schematically in Fig. 38. Once the primary independent variables (i.e., fuel type, power plant design generation, thermodynamic cycle, number of compressor spools, turbine cooling scheme, turbine inlet gas temperature, compressor pressure ratio, and power plant capacity) have been specified in the program, all combinations of the design parameters listed in Table 59 (incorporated into the program) are investigated, and the appropriate combination of parameters that satisfy the design constraints denoted by asterisks in Table 59 are determined. The power plant flow path, pertinent dimensions, number of compressor and turbine stages, turbine cooling flow requirements, and allowable metal temperatures are then computed. After the aforementioned parameters have been determined, the computer program assesses the effect of the calculated turbine cooling flow rate, including the collective contributions of blade and vane cooling flows and disk cooling flow, on specific power and thermal efficiency.

Constant mean diameter flow passages were assumed for all turbomachinery performance computations in order to provide some control over the number of possible design concepts. The number of compressor stages in each spool was computed on the premise of a constant flow coefficient per stage. This assumption together with the preceding one concerning constant mean diameter results in a constant axial velocity and, hence, constant stage work for each of the respective compressor spools. However, for twin-spool compressor designs, the mean diameter of the low compressor was not required to be equal to that of the high compressor; thus, each spool was allowed to operate at its maximum stage efficiency. The high-pressure turbine for all power plant designs was assumed to comprise a single stage, capable of delivering up to the maximum stage work levels dictated in Table 59. Supplementary compressor work requirements were divided equally among high-efficiency, low-pressure turbine stages whose number was determined from the appropriate design parameters and constraints listed in Table 59. Designing for high-stage work and hence a relatively high temperature drop in the high turbine reduces the cooling flow requirements and possibly the high-temperature materials requirements in the low-pressure turbine.

Turbine blade- and vane-cooling flow requirements were obtained from correlations of cooling effectiveness, (η), with cooling flow for both advanced impingement-convection cooling techniques and transpiration cooling techniques (Ref. 107). Cooling effectiveness is defined as the difference between average blade or vane metal temperature and cooling air temperature divided by the difference between gas temperature and cooling air temperature. The correlations for advanced impingement-convection cooling are comparable to the best current cooling designs for aircraft gas turbine propulsion systems. The correlations for transpiration cooling were derived by extrapolation of experimental data. Disk cooling flow requirements were assumed on the basis of related engineering experience, since accurate determinations of this parameter would have entailed detailed heat transfer and seal leakage analyses.

The average blade metal temperatures used to determine cooling effectiveness, defined in the preceding discussion, were computed with the aid of creep strength data (1% creep in 100,000 hr) shown in Fig. 32 used in conjunction with a simplified blade root stress relationship, derived in Ref. 107. In the use of this data, emphasis was placed on the best material available during the specified time period to minimize cooling flow requirements, as an alternative to the use of less expensive alloys with commensurately higher cooling flow requirements. This design philosophy was also used to determine the power turbine performance. As a result, none of the power turbines designed in this study required cooling. A maximum allowable blade root stress of 40,000 psi was imposed for the first- and second-generation designs, and 47,000 psi was imposed for the third-generation designs.

Some sources (e.g., Ref. 105) report that the creep life of a cooled turbine blade, during steady-state operation, will be different from that of an uncooled blade whose creep life is simply a function of the blade material and of the imposed temperature and stress levels. In a cooled blade the chordwise differences in metal temperature result in thermal stresses which can be several times as great as the direct centrifugal stress. The result is to produce high tensile stresses in the cooler parts of the blade (the core) and low tensile or even compressive stresses in the hotter parts of the blade (the leading and trailing edges). It is also reported in Ref. 108 that the blade creep life taking account of thermal stress effects has, in systems experience, worked out to be approximately one-fifth of the creep life that would be calculated on the basis of average chordwise temperature and stress distribution. This would imply that cooled blades, particularly those employing single-cavity convection cooling configurations, might have a useful life of 20,000 hr rather than the 100,000 hr life-time goal set for the complete power plant. It is expected, however, that the useful creep life of cooled industrial turbine blades can be improved substantially with the development of more sophisticated blade cooling configurations such as the impingement-convection type considered in this study.

PERFORMANCE CHARACTERISTICS OF FUTURE BASE-LOAD GAS TURBINE POWER PLANTS

Projected improvements in base-load gas turbine thermal efficiency and specific output (shp/unit airflow) performance based on predicted advances in thermodynamic cycle, component efficiencies, turbine cooling schemes, and high-temperature alloys, encompassing three generations of design technology and covering the time period from the present through the 1990's are described in the following paragraphs. The performance was computed only for ambient conditions defined in accordance with NEMA standards; namely, 80 F and 1000 ft above sea level.

Parametric Performance Analysis for Simple-Cycle Gas Turbines

Thermal efficiency and specific output, based on present-day design technology, are depicted in Figs. 38 and 39, respectively, for hypothetical Fuel Gas I-fired, simple-cycle power plants with turbine inlet gas temperatures ranging from 1600 to 2200 F and compressor pressure ratios from 12 to 38. The performance levels depicted in these figures are based on designs incorporating advanced impingement-convection turbine blade and vane cooling configurations that utilize cooling air bled from the compressor exit and delivered to the turbine at compressor discharge temperature and pressure. An inspection of Figs. 39 and 40 shows that although a peak value of thermal efficiency of 35.5% would be achieved at a turbine inlet gas temperature of 2000 F, specific output continues to increase with further increases in gas temperature. At this 2000 F temperature level, the optimum compressor pressure ratio for maximum thermal efficiency performance would be on the order of 18 to 22.

Further improvements in thermal efficiency with additional increases in cycle conditions (temperature and pressure) would not be achieved because of the increasing influence of turbine cooling flow penalty on power plant performance. This trend becomes evident from an inspection of Figs. 41 and 42, respectively, which depict the degradation in thermal efficiency and specific output with turbine cooling flows. Increases in design pressure ratio can contribute to higher turbine cooling flow requirements in two ways; first through the need for additional turbine stages, and secondly through a reduction in blade cooling effectiveness caused by an increase in compressor discharge (coolant) temperature. This second consideration results, as shown in Figs. 39 and 40, in a limit in the maximum compressor pressure ratio that is achieved with a particular turbine cooling scheme.

Providing supplementary cooling of the compressor discharge bleed flow prior to its use for turbine cooling increases the design cooling effectivenss and allows further increases in turbine inlet gas temperature and/or compressor pressure ratio with advanced impingement convection cooling techniques. The dashed curves superimposed on Figs. 39 and 40, respectively, depict the effect on thermal efficiency and specific output performance of using compressor discharge bleed flow precooled to two different temperature levels (250 F and 125 F) for turbine cooling purposes.

Parametric curves depicting base-load performance for first-generation simple-cycle, Fuel Gas I-fired gas turbine power plants are summarized in Fig. 43. Three bands of data, each reflecting different degrees of turbine cooling, are presented. The upper limits of gas turbine performance shown in Fig. 43 are based on the assumption of uncooled materials capable of withstanding the specified gas temperatures. These curves obviously represent fictitious power plant designs, since at this time materials capable of withstanding these extreme temperatures are not projected to become available within the first-generation time period. Similar plots summarizing base-load performance that can be expected to be achieved during the next two decades are presented in Figs. 44 and 45. This improved performance is based primarily on the use of the improved high-temperature materials described in Fig. 4 which, with the use of advanced impingement-convection cooling schemes currently developed, will allow the attainment of the high turbine inlet gas temperatures shown on Figs. 44 and 45. Also contributing to these high performance levels, however, are the improvements in turbomachinery component efficiencies documented in Table 58. Performance penalties that would be imposed by the turbine cooling flows on these second- and third-generation technology power plant designs are depicted in Figs. 46 through 48.

Parametric curves depicting base-load performance for first-, second-, and third-generation design technology gas turbine power plants that would burn a representative high heat-content (HHV = 1000 Btu/ft^3) natural gas fuel are presented in Figs. 49 through 51, respectively. It becomes evident from a comparison of the performance depicted in these figures with those for power plants, burning low-heat-content Fuel Gas I, (Figs. 43 through 45) that considerably higher power plant performance could be realized with the combustion of the low-heat-content fuel than with the combustion of high-heat-content natural gas fuel. The reasons for this trend are due primarily to the accounting procedures that were used in assessing performance penalties for the auxiliary power required to pressurize the fuel to desired combustor pressure levels. It was assumed for both fuel types that the cost for fuel pressurization would be assessed to the fuel processing system rather than to the power plant since the efficiencies of the coal gasification processes investigated in Part One improved at high system reaction pressures. However, since the required fuel flows for the low-Btu designs can be greater than the fuel flows for the natural gas-fueled designs by a factor as great as ten, the availability of this high-pressure gas at no cost in performance to the power plant proves highly beneficial in net power plant output capabilities.

It can be seen from an inspection of Fig. 52 that power plant designs employing proven advanced impingement-convection cooling schemes could, if they utilize supplementary cooling of the compressor discharge bleed air, achieve thermal efficiency and specific output performances nearly the same as those predicted for power plant designs incorporating the transpiration cooling schemes.

Parametric Performance Analysis for Regenerative-Cycle Gas Turbines

Thermal efficiency and specific output performance of hypothetical regenerative-cycle gas turbines reflecting advances in design technology projected over the three time periods of interest are summarized in Fig. 53 for the low-heat-content gas-fueled power plant designs and in Fig. 54 for the high-heat-content natural gas-fueled designs. These performance levels were computed for a regenerator total pressure drop of 4.0%, equally divided between the air side and gas side (Ref. 109).

The performance data depicted in Fig. 53 are based on the utilization of supplementary cooling to a level of 250 F for the compressor discharge bleed flow prior to use of this air in cooling the turbine section. A comparison of this data with comparable data in Figs. 43 through 45 for the simple-cycle configurations clearly shows that regeneration does not provide any significant improvement in the thermal efficiency or specific output of power plants employing low-heat-concent fuels. In fact, the relatively high turbine inlet temperature power plant designs (based on second- and third-generation design technologies) employing regeneration actually show a poorer performance than the simple-cycle designs. These trends can be explained by the fact that the high fuel-to-air ratios associated with the power plants burning this low-heat-content fuel result in the recovery of only a relatively small fraction of the available heat in the exhaust gases; hence, the temperature rise in the compressor discharge air is not enough to provide a significant reduction in fuel flow rates required to achieve a specified turbine inlet gas temperature.

An inspection of Figs. 50 through 52 shows, as would be expected, that by comparison with simple-cycle designs the regenerative-cycle power plants utilizing high-heat-content fuel exhibit a small but significant improvement (3 to 4 percentage points) in thermal efficiency performance. This improvement in thermal efficiency is also accompanied by an improvement of as much as 10% in specific output, due to the relatively low compressor pressure ratios associated with the regenerative-cycle designs. However, in order to realize these improvements in performance, a regenerator air side effectiveness of at least 80% is required. Inlet and exit gas temperatures associated with 80% air side effectiveness regenerator designs are depicted in Figs. 55 through 57. It is apparent from the temperature levels indicated in these figures that relatively expensive high-temperature heat transfer materials would be required for the regenerators. In the final analysis, justification for the utilization of regenerative-cycle systems employing high-heat-content fuels must be made on an economic basis.

Gas Turbine Part-Load Characteristics

In nondimensional form the part-load performance characteristics for the advanced gas turbine engines considered in this study would exhibit the same general trends illustrated in Fig. 58 for present-day engines. However, some differences should be expected due to the higher compressor pressure ratios and turbine inlet temperatures utilized in the advanced engines. It may be noted that the correlation in Fig. 58 (taken from Ref. 95) is for two-shaft (free turbine) gas turbines operating at full-load (constant) output speed and that the advanced study engines are free turbine engines which must operate at constant speed in the intended electric power application. The correlation in Fig. 58 is represented as a wide band especially at the lower part-load settings because of the scatter in the original data points. The scatter is due to the differences in engine designs (pressure ratio, turbine inlet temperatures, etc.) represented.

If high performance at part-load power were desired, the power turbine could be equipped with variable inlet guide vanes, but these would add to the complexity and cost of the engine. In most gas turbine engines, the turbine inlet guide vanes are fixed, and part power is achieved by reducing the turbine inlet temperature. Although the airflow drops slightly at part power, the lower work capability of the working fluid due to the reduction in turbine inlet temperature is the primary cause of the power decrease, and it is well known that gas turbine performance decreases significantly with lower turbine inlet temperatures. Part-load power can be achieved at constant turbine inlet temperature by utilizing variable-geometry turbine inlet guide vanes which have choked flow and therefore control the airflow through the engine. Although some degradation in component performance results from the reduced airflow and changes in the turbine inlet guide vane angle, the part-load performance of a gas turbine thus equipped would approach the full-load performance value. Naturally, there is a limit to the reduction in airflow which could be achieved with variable-geometry turbines because of mechanical considerations. At some point it would not be desirable to reduce output power by this method because of turbine efficiency considerations.

The part-load characteristics of the gas turbine are of secondary importance in this study because the power station design load factor selected requires the engine to be base-loaded, i.e., operating most of the time at rated power. Furthermore, to analytically determine the part-load performance of a gas turbine requires the use of a relatively complex engine matching procedure and involves a detailed knowledge of the off-design performance of each component. Unless representative off-design performance maps for the compressor and turbine components are available, the off-design performance of these components must be estimated by analyzing the velocity triangle diagrams of each stage. Thus because of its relative unimportance and the complexities involved, the part-load characteristics of gas turbine-engines were not investigated in detail in this study.

CHARACTERISTICS OF SELECTED BASE-LOAD GAS TURBINE POWER SYSTEMS

The potential performance of advanced-cycle gas turbines is well above that of currently available gas turbines and in fact would be competitive with both conventional fossil-fuel steam stations and some of the closed-cycle power systems investigated during this study. The selection of the three generations of gas turbine power systems and estimates of the busbar power costs are discussed below.

Design Characteristics

A summary of the design specifications for first-, second-, and third-generation gas turbines for use in a base-load power station is given in Table 60 These specifications include the number of units per 1000-Mw station, operating conditions, performance, and general dimensions for the units.

All three generations of gas turbines would have two spools and a separate power turbine operating at 1800 rpm. The gas turbine inlet temperatures were taken to be 2200 F, 2800 F, and 3100 F and pressure ratios were 20, 32, and 34 for first-, second-, and third-generation designs, respectively.

Performance Characteristics

Selected performance data also presented in Table 60 indicate the gas turbine unit output and projected power system efficiency. The net system efficiencies are shown to be 35.2%, 41.5%, and 43.6% for the three generations. These efficiencies are comparable to those of some other advanced-cycle power systems and those projected for future coal-fired stations.

It should be noted that all of the heat rejected from the base-load gas turbine power systems would be rejected directly to the atmosphere; thus, there would be no thermal pollution of cooling water for this type of power station.

Economics

The gas turbine design data presented in Table 60 form the basis of power system cost estimates presented in Tables 61 through 66. These cost estimates are tabulated according to standard FPC accounts which include land and land rights, structures and improvements, prime movers (gas turbines), electric generators, accessory electrical equipment, miscellaneous power plant equipment, and miscellaneous station equipment. Capital costs for the gas turbines were estimated using the gas turbine costing procedure detailed in Appendix G. All other equipment cost estimates are based upon detailed cost-estimating procedures used to estimate costs for conventional generating stations (see Appendix D). A summary of station cost estimates is given in Table 67 for the three generations of power systems. As shown in this table, the total direct and indirect costs are estimated to be approximately \$77.5 million, \$60.8 million, and \$61.7 million, respectively, for the three generations of power systems.

Operating and maintenance costs for the gas turbine power systems were assumed to be 0.5 mills/kwhr of net gas turbine output. Supplies and materials were estimated to cost approximately 0.2 mills/kwhr of net power system output.

The estimated power generation costs for the three generations of gas turbine systems are presented in Fig. 59 in terms of the total cost of fuel delivered to the power systems. According to the data presented in Part One, the total cost of fuel delieverad to the first-, second-, and third-generation gas turbine power systems from fuel conversion processes incorporating the partial oxidation of coal are estimated to be approximately 40.4¢/million Btu, 31.4¢/million Btu, and 31.8¢/million Btu, respectively. The busbar power costs, based on these fuel costs, are estimated to be approximately 7.9 mills/kwhr, 5.4 mills/kwhr, and 5.4 mills/kwhr, respectively. While the first-generation system has busbar costs approximately 1.7 mills/kwhr greater than a conventional coal-fired steam station, the projected busbar costs for the later generations are significantly less than those projected for comparable conventional coal-fired stations. The third-generation power cost shows no significant improvement over that of the second-generation because of two factors: (1) slightly higher third-generation power system capital cost, and (2) the power requirement for the tail-gas recycle compressor for the third-generation sulfur removal system.

Section III - Combined Gas Turbine and Steam Turbine Power Systems

This section contains estimated design, performance, and economic characteristics of prospective combined gas turbine and steam turbine (COGAS) power systems incorporating large industrial-type gas turbines. Numerous past studies have been concerned with the adaptation of existing gas turbine technology to COGAS system design (see Refs. 110 through 115, for example), but it has been only comparatively recently (Refs. 116 and 117) that the future high-performance potential of COGAS systems has been recognized. The present study involved incorporation of advanced design features derived from aircraft gas turbine engines into advanced-cycle, high-performance COGAS power systems. An analysis was conducted to select the COGAS system configuration which looked most promising for future commercial applications. Performance estimates were based on a high-speed digital computer program developed to facilitate computation of gas turbine and steam system output powers, fuel consumption, steam cycle heat balance, power system efficiency, and other useful thermodynamic quantities. Detailed capital cost tabulations are presented for 1000-Mw nominal output base-load COGAS power systems, and busbar power costs are estimated in terms of the net cost of fuel delivered to the power system. It appears, from the results of these analyses, that with the advent of higher gas turbine inlet temperatures, made possible by the results of advanced aircraft development programs and improved turbine blade cooling techniques, advanced-cycle COGAS power systems will show significant improvements in thermal efficiency with time and will merit serious consideration for future electric power generation.

DESCRIPTION OF COGAS POWER SYSTEMS

Five basic COGAS power system configurations were evaluated during the present investigation. They are exhaust-fired, waste-heat recovery, conventional supercharged, gas generator supercharged, and two-pressure supercharged systems. These five basic configurations encompass all COGAS configurations of interest. Many variations of these five basic configurations involving compound gas turbine cycles with intercooling, reheat, and regeneration were also investigated. In addition, several different steam system feedwater heating cycles were investigated. The principal characteristics of these five basic COGAS system configurations are described in the following paragraphs.

Exhaust-Fired System

The only large-capacity, base-load COGAS power systems that have been installed by US electric utilities are of the exhaust-fired type. Three large-capacity systems of this type with outputs ranging from 85 to 250 Mw have been installed in the US southwest. Several utilities are currently entertaining proposals for exhaust-fired COGAS systems for peaking and intermediate-load applications. A schematic diagram of the exhaust-fired system configuration is given in Fig. 60. In this system, air would be compressed, fuel injected into the gas turbine burner and burned, and the products of combustion expanded through the compressor-drive turbine and the power turbine which drives the electric generator. The power turbine exhaust gases would be passed into the steam boiler where additional fuel would be injected and burned. Exhaust-fired boilers are nominally low-pressure boilers because the gas pressure is only slightly above atmospheric pressure (approximately 20 to 50 in. H_2O gauge). The final excess air in the combustion products exhausted through the stack would vary from 5% to over 100%, depending on the desired power split between the gas turbine and steam system.

Only the main steam and feedwater flows are depicted in the simplified steam cycle illustrated in Fig. 60. If complete firing were employed in the boiler, corresponding to low excess air in the stack gases, the steam cycle would be similar to the high-efficiency cycles with 7 or 8 stages of regenerative feedwater heating utilized in conventional base-load steam stations. If minimal boiler firing were employed, corresponding to relatively high excess air in the stack gases, the steam cycle would be considerably simplified with only 0 to 3 stages of feedwater heating.

There are two serious disadvantages of exhaust-fired systems: (1) increased complexity of fuel firing and control in the boiler, and (2) degradation of steam cycle performance. There are also several secondary disadvantages which are described in Refs. 112, 118, and 119. The problem with fuel firing in the boiler results from the fact that gas turbine exhaust gases, which would be utilized as preheated combustion air for the boiler, would be much hotter than the 550 F to 650 F temperature of the preheated combustion air normally supplied to conventional boilers. Present-day gas turbines have exhaust gas temperatures in the 750 to 900 F range, and future gas turbines will have even higher exhaust temperature. As a result of this higher gas temperature, the volume flow of combustion gas would be larger; the cooling effect of the gas on the front part of the burner, which would be exposed to flame radiation, would be lessened; movable burner parts, such as control vanes, would be more likely to malfunction because of warping or inadequate lubrication; and control of thermal expansion and resulting relative displacement of furnace walls, burner parts, and wind boxes would become more difficult. These problems would become more serious as boiler firing is increased. In addition, the gas flow rate delivered by the gas turbine would be practically constant at all loads, whereas the fuel flow to the burners in the boiler would be reduced according to load. Feeding the entire gas turbine exhaust flow into the burner wind box at all loads means that the fuel/air ratio of the burner must vary greatly, which would add to the difficulty of burner design. Therefore, part of the gas turbine exhaust flow must be bypassed around the burner wind box at part loads and either passed directly into the stack through boiler bypass ducting (which would be detrimental to plant efficiency and require control dampers, in addition to isolating dampers, in the bypass line) or fed into the furnace through a separate burner bypass line (which would further complicate the system of ducting). As a result of these complications in burner and bypass design and the need for burner materials suitable for higher temperatures, the burners and controls for boilers designed for exhaust-fired combined systems would be appreciably more expensive than those for conventional steam boilers.

The degradation in efficiency of the steam cycle would result from the inability to incorporate an air preheater to cool the approximately 600 F combustion gases exiting from the economizer to 300 F or below. The air preheater must be eliminated because the temperature of the combustion air discharged from the compressor would be much to high too be used to cool the stack gases. Instead, a stack gas cooler (SGC), sometimes referred to as a primary economizer, would be used to recover the energy in the low-temperature combustion products, as shown in Fig. 60. As described in Ref. 114, the heat recovered in a SGC would be converted into power at an efficiency of only 60%, approximately, of steam cycle efficiency, whereas the heat recovered in an air preheater would be converted into power at full steam cycle efficiency. The lower energy conversion efficiency results because the amount of steam extracted from the steam turbine for feedwater heating must be reduced to reflect the feedwater heating accomplished in the SGC. This reduction in steam extraction means that there would be less regeneration in the steam cycle, and, therefore, the thermal efficiency of the steam cycle would be decreased.

The degradation of steam cycle efficiency due to the reduction in steam regeneration, however, would be more than offset by the increase in overall power system efficiency due to combining the steam system with the gas turbine (see subsequent performance analysis). As a result, the thermal efficiencies of exhaust-fired COGAS systems would be higher than those for conventional steam systems. Additional gains would be realized in COGAS systems through a reduction in auxiliary power requirements brought about by elimination of the forced-draft and induced-draft fans from the boiler circuit and by the reduced boiler feed pump power (for a given total power system output power).

As illustrated in Fig. 60, the feedwater flow would be split after the boiler feed pump. Part of the feedwater would flow through the high-pressure feedwater heaters to the economizer and part would flow through the SGC to the economizer. Highest efficiency for this type of system would be achieved when the temperatures of the two feedwater streams are equal before the streams are joined and passed into the economizer. The particular feedwater heating configuration depicted in Fig. 102 is referred to as a parallel-flow heater cycle because the SGC and high-pressure feedwater heaters are in parallel. Alternate configurations are the series-flow heater cycle in which all the feedwater passes through the SGC before passing through the high pressure feedwater heaters and the short heater cycle in which the high-pressure feedwater heaters are eliminated and the SGC is an integral part of the economizer. The efficiencies of each feedwater heater cycle decrease in the order of discussion, but their capital costs also decrease in the same order so that the economically optimum feedwater configuration for any particular application would depend on the fuel cost. Since clean fuel for COGAS systems will be very expensive (30 to 50¢/million Btu), a greater premium was placed on operating efficiency than first cost and the parallel-flow feedwater heating cycle was assumed for all COGAS systems incorporating regenerative feedwater heaters.

Waste-Heat Recovery System

Waste-heat recovery COGAS systems (see Fig. 61) are essentially degenerate forms of exhaust-fired systems without supplementary firing in the steam boiler. Small-capacity heat-recovery systems are common in industrial applications and have also been used in some peaking and intermediate-load utility applications. Plans have recently been announced (Ref. 120) for erection of a 300-Mw waste-heat COGAS plant in a large industrial complex. The four gas turbines in this plant would be fired with gas, operate at 1600 F to 1700 F turbine inlet temperature, and produce approximately 200 Mw or two-thirds of the total system output. The efficiency of this COGAS system is estimated to be approximately 36%, which compares favorably to conventional gas-fired steam stations.

The primary advantages of future waste-heat systems utilizing advanced-design gas turbines would be their very high performance (see subsequent discussion of performance) and relative simplicity of design compared to other COGAS systems. For example, steam cycles for waste-heat systems, as illustrated in Fig. 61 would not employ regenerative feedwater heating, and the heat content of the gases entering the boiler economizer (which contains the SGC), would be used to preheat the feedwater. Also, since boiler firing would not be employed, complex firing and fuel controls would not be needed for the boiler. Similarly, all the other boiler problems previously described for exhaust-fired systems would be eliminated. Furthermore, since the steam system could not operate separately from the gas turbine (a slight disadvantage), there would be no need to install forced- or induced-draft fans and drive motors, dampers, controls, and other equipment required to operate an exhaust-fired boiler independently of the gas turbine. In addition, boilers for advanced-design waste-heat systems would be designed for forced-convection heat transfer with higher pressure drops (approximately 40 to 100 in. H_2O) than commonly utilized in present-day waste-heat systems (8 to 20 in. H_2O) leading to much more compact boiler design. This simplicity and compactness of design should result in substantially less expensive construction costs than for other COGAS systems.

Another major advantage of waste-heat COGAS systems is that gas turbines designed for this application would also be optimum for gas turbine peaking applications. Engines for both applications would basically be designed for low pressure ratio, high specific output, and low cost. Gas turbines designed for straight base-load applications, on the other hand, would be designed for higher pressure ratio, higher efficiency, and higher cost. Obviously, considerable development cost savings would be possible if a single gas turbine could be developed for both peaking and base-load COGAS applications.

The primary disadvantage of waste-heat COGAS systems is the substantial degradation in steam cycle performance which would result from complete elimination of the feedwater heating cycle. Whereas the efficiency of the steam cycle in an exhaust-fired system might be degraded by 2% to 4% (for complete firing in the boiler) compared to a conventional steam cycle, the efficiency of a waste-heat steam cycle could be as much as 13% to 15% less than the efficiency of a conventional steam cycle. Thus, the efficiency of a 2400 psig/1000 F/1000 F steam cycle would be reduced from approximately 45% in conventional steam system applications to approximately 39% in waste-heat applications. However, as discussed for exhaust-fired and other COGAS cycles, the degradation of steam cycle efficiency would be more than offset by combining the two cycles. A detailed discussion of waste-heat steam cycles and selection of optimum steam conditions is presented in a subsequent discussion of waste-heat recovery COGAS system design considerations.

Supercharged System

In the supercharged COGAS system, Fig. 62, air from the compressor would be discharged directly into the high-pressure (12 to 16 atm) furnace of the boiler wherein all combustion would take place. The resulting combustion products would contain from 5% to over 100% excess air, depending on the desired power split between the gas and steam systems. The high-pressure combustion products would be cooled in the high-pressure section of the boiler until the gas temperature is low enough to handle in the gas turbine. The combustion products would then be expanded through the compressor-drive and power turbines and directed through the low-pressure (20 to 50 in. H_2O gauge) economizer and SGC for feedwater heating.

If the net power output from the power turbine is zero, as would be the case when the cycle pressure ratio is less than approximately 5 (depending on turbine inlet temperature), then the system would be similar to the Velox supercharged systems introduced in Europe in the 1930's by Brown Boveri and Co., Ltd. Velox systems were the first type of COGAS system to be utilized in commercial applications. Although approximately 115 of these units have been sold in Europe since the 1930's, none were sold in the US until recently when Velox-type propulsion units were installed in 10 US Navy 1040-class destroyer escorts.

The output from a supercharged system can be greater than that from an exhaust-fired system for a fixed compressor airflow because of the higher output from the gas turbine in the supercharged system. This higher output results from higher mass flow through the gas turbine (due to higher fuel/air ratio of gases expanded through the turbine) and from the higher sensible heat in the steam formed during combustion. Other advantages of supercharged systems are their apparently lower first cost and associated compact boiler design afforded by pressurization of the boiler. These latter advantages are a direct result of the fact that supercharging a boiler increases the Reynolds number for gas flow and, therefore, increases the convective heat transfer coefficients on the hot gas side. Gas radiation also increases with pressure, although to a lesser degree, and flame dimensions diminish. As a result, the required heat transfer surface area (see Fig. 63) as well as the furnace volume become smaller, thus reducing physical size and weight. The reductions in size and weight apparently permit substantial cost reductions as described by one boiler manufacturer in Refs. 121 and 122. In Ref. 121, it was stated that supercharged boilers designed for 5 and 8 atmospheres should cost only 57% and 49%, respectively, as much as conventional atmospheric-pressure boilers designed for the same gross output. In extending pressure from 8 atm to 12 atm and above, diminishing returns would be experienced regarding heat transfer surface area and cost reductions. However, the physical dimensions achieved for the pressure parts for a 400-Mw boiler, particularly at 12 atm and above, bring into focus a possible module-type shop construction technique for large-output boilers which would reduce field erection costs (Ref. 122).

The principal disadvantage of a supercharged system is that it would require a boiler design which is radically different from that of a conventional steam boiler - a new design replete with new development problems which would need careful investigation. The results reported in Refs. 118 and 121, indicate that technical solutions to these problems do exist, but domestic boiler manufacturers do not appear to be inclined toward large-scale boiler development programs.

The degradation in the performance of the steam cycle in a supercharged system due to eliminating the air preheater and reducing the amount of regeneration in the steam cycle would be identical to the performance degradation in an exhaust-fired system. Another disadvantage of supercharged systems is that the steam system could not be operated independently of the gas turbine should one of the gas turbine components fail. This consideration is expected to reduce the reliability and availability of supercharged systems relative to conventional steam systems, but no quantitative assessment of this reduction has been made.

Gas Generator Supercharged System

The gas generator supercharged system (the two-pressure type of supercharged system described subsequently) is an unconventional type of COGAS system which has never been used in commercial applications. As may be seen from the schematic diagram in Fig. 64, the gas generator supercharged system basically combines the conventional exhaust firing concept with the supercharging concept. A gas generator, consisting of an air compressor, burner, and compressor-drive turbine, would be used to supercharge the boiler. The exhaust from the gas generator would be passed into the boiler, as in the exhaust-fired case, but the boiler would be at moderate pressure level (3 to 8 atm) instead of at high pressure as in the conventional supercharged system. Additional fuel would be burned in the boiler. After passing through the boiler the combustion products, still at moderate pressure, would be expanded through the power turbine. From the power turbine the gases would be passed through a low-pressure SGC, where feedwater would be heated, and finally exhausted through the stack.

Despite degradation of the performance of the steam cycle, which is common to all combined systems, the gas generator supercharged system would also be capable of higher thermal efficiency than conventional steam systems. Its efficiency, however, would be less than for other COGAS systems, as indicated in a subsequent discussion on comparative performance. Another major advantage would be the compact, low-cost boiler design afforded by pressurization to the 3 to 8 atm range. The advantages in this regard are similar to those previously described for the conventional supercharged system.

The principal disadvantage of this type of system is the increased complexity of fuel firing and controls resulting from passing the very hot (over 1500 F) exhaust gases from the gas generator through the burners into the furnace. This disadvantage and all of its ramifications are similar to those described for conventional exhaust-fired systems, although to a greater degree, and are not discussed further.

Two-Pressure Supercharged System

A schematic diagram of the unconventional two-pressure supercharged system configuration is presented in Fig. 65. In this system air would be compressed to high pressure (12 to 16 atm) and discharged directly into the furnace of the boiler. All the fuel to be burned would be injected into the furnace. The high-pressure combustion products would be passed through the furnace, and, if necessary, passed over superheater and reheater surfaces in the boiler until the gas temperature is reduced enough to expand the gases through the compressor-drive turbine. Thus far, operation of the two-pressure supercharged system would be identical to that for conventional supercharged systems. The moderate-pressure (3 to 8 atm) exhaust from the compressor-drive turbine, however, would be returned to the boiler. From this point on, operation would be identical to that for the gas generator supercharged system.

The two-pressure supercharged system would be capable of higher thermal efficiencies than the gas generator supercharged system, as subsequently described, while not requiring the same type of complex fuel control and hot gas bypass systems which would be required if additional fuel were burned in the boiler.

A more significant advantage would result from the fact that the furnace could be separated from the economizer and superheater sections of the boiler. These latter sections require large heat transfer surface areas, and the data in Fig. 63 suggest that the optimum gas pressure for sections requiring large area would be in the range of 3 to 5 atm. Cost estimates presented in Refs. 121 and 122 support this conclusion. The furnace, on the other hand, requires large volume to prevent flame impingement on the walls and to achieve complete combustion. In conventional atmospheric-pressure, gas-fired steam generators, approximately 50% of the total volume (but less than 10% of the total heat transfer surface area) is required by the furnace. Since both gas specific volume and flame dimensions diminish with increased pressure, there is greater incentive to increase furnace pressure than to increase pressure in other boiler sections. Preliminary indications are that a furnace pressure of 12 to 16 atm, when coupled with the remaining sections of the boiler operating at 3 to 8 atm, would result in the lowest boiler cost.

ANALYSIS OF ADVANCED-CYCLE COGAS POWER SYSTEMS

The following paragraphs summarize the basic assumptions, definitions, and method of analysis used to derive COGAS power system performance estimates. In addition, a brief discussion is presented which incorporates both preliminary performance and capital cost estimates and leads to selection of the waste-heat recovery type of COGAS power system as being the COGAS system type which offers the greatest future potential for low-cost power generation. Finally, waste-heat recovery COGAS power system performance trends and part-load operating characteristics are presented.

Assumptions

Although Part Two of this report does not deal directly with that portion of the power station which must supply fuel to the power system, it is appropriate to briefly discuss the general characteristics of the type of fuel which would be used in COGAS power systems. As discussed in Part One of this report, the most economical method of obtaining low-sulfur fuels which meet prospective sulfur oxide emission control standards appears to be partial oxidation of coal or residual oil in the presence of steam and air at elevated temperature and pressure. The product gas from this process, with sulfur in the form of H_2S and COS, would be sent to a desulfurization unit which would remove the undesirable sulfur compounds. The gasification processes and desulfurization techniques of interest would typically operate at elevated pressures and deliver a high-pressure fuel gas having a heating value of 150 to 200 Btu/ft^3 at standard conditions. Fuel Gas I, a representative low-Btu fuel gas (see Appendix F), was used to establish the performance trends described subsequently, although some early performance comparisons based on the use of methane are also described. Performance estimates for the integrated COGAS system designs presented in Part Three are based on Fuel Gas II and Fuel Gas III.

All design and performance estimates presented herein are based on power systems designed exclusively for high-load factor, base-load operation, i.e., it was assumed that whenever the power station would be in operation it would produce its rated output. No compromises in design, performance, or capital cost of the final power systems were included to permit these power systems to operate efficiently at reduced output, although a subsequent discussion presents general part-load performance trends and indicates typical gas turbine design changes which would be necessary to permit high-efficiency operation at part-load.

The anticipated advances in design technology and materials improvement projections presented in Part Two, Section II formed the basis of gas turbine performance incorporated in parametric COGAS system performance studies. Gas turbines designed for straight base-load and COGAS applications would not be the same, i.e., the compressor pressure ratio, number of spools, number of compressor and turbine stages, etc., would be different for each application. However, the gas turbine aerodynamic flow path design, high-temperature material selection, blade cooling techniques, and other design technologies for COGAS applications would be basically the same as described for base-load gas turbines and is not repeated. Projections for some of the more important gas turbine design parameters reported in the section on base-load gas turbines are summarized in Table 68.

Performance estimates of steam systems for COGAS applications were based on steam turbine and boiler data collected and correlated by UARL. Detailed design and performance data for waste-heat steam systems is presented in the following section on system design considerations. The gas-side pressure drops assumed for each section of exhaust-fired and supercharged steam boilers are tabulated in Table 69. The gas-side pressure drops through waste-heat boilers were varied from 2% to 50% of gas pressure at the inlet of the boiler in order to determine the pressure drop which would result in lowest electric power generation cost. The power system auxiliary power requirements were assumed to be 0.3% of the gas turbine net output plus 1.9% of the steam system net output. This steam system auxiliary power requirement is 2.8 percentage points less than the 4.7% auxiliary power requirement which is typical of conventional gas-fired steam stations because the forced-draft and induced-draft fans normally required for conventional steam boilers would not be required in COGAS systems.

Efficiency Definitions

The steam cycle efficiency, as used herein, is defined as the ratio of steam turbine ouput (including boiler feed pump power consumption but excluding other steam system and boiler auxiliaries) divided by the total heat transferred into the steam and feedwater in all sections of the boiler. Stack losses are not included in this definition. Power system efficiency calculated by the method described in the following paragraphs does include stack losses, however, and is defined herein as the power output from the COGAS power system (after subtraction of all power system auxiliaries) divided by the total chemical plus sensible heat delivered to the power system. By this definition, power system efficiency does not include thermal losses in the fuel conversion system. Final performance estimates for COGAS power stations presented in Part Three of this report do include all fuel conversion losses.

Method of Performance Analysis

The analytical method used to estimate the performance of exhaust-fired and supercharged COGAS systems is based on a procedure described in Ref. 123. This procedure consists of dividing the COGAS system into three sections consisting of the steam generator, steam turbine system, and the load gas turbine or power turbine. The power output from the COGAS system is grouped into three distinct blocks of power, each with its own efficiency. The three blocks of power are: power from the base steam system (before modifications are made to the feedwater cycle to accommodate the SGC), incremental steam system power (which results from the increased steam flow through the steam turbine due to reduced steam extraction for feedwater heating), and gas turbine power. The Ref. 123 procedure presents a method for estimating the incremental steam turbine output and its heat rate for any selected base steam system. The reader is referred to Ref. 123 for details of this procedure, but an outline of the procedure consists of the following steps: (1) calculate a heat balance about the components designated as the steam generator equivalent (see next paragraph); (2) calculate the heat input to the SGC, the resulting incremental steam turbine power, and the efficiency of this incremental power; (3) calculate the output of the power turbine; (4) calculate the total gross COGAS system output and efficiency; and (5) apply auxiliary power and boiler losses to determine net COGAS system output and overall efficiency.

In calculating the performance of COGAS systems, it is convenient to first define an equivalent boiler for exhaust-fired and supercharged COGAS systems. An example boiler equivalent is illustrated in Fig. 66 for the conventional supercharged system. The compressor and compressor-drive turbine are regarded as part of the boiler equivalent and as replacements for the forced-draft fan and fan drive, respectively. The power turbine-generator set illustrated in Fig. 66

is regarded as producing power with only mechanical and generator losses, since the heat from the turbine exhaust would be recovered in the low-pressure sections of the steam boiler. With an equivalent boiler as defined in Fig. 66, a boiler efficiency can be defined which is comparable to the efficiency of a conventional steam boiler; namely, unity minus the summation of losses due to moisture formed during combustion, moisture in the air, moisture in the fuel, sensible heat in the stack gases, unburned combustible material, radiation from boiler walls, and unaccounted for losses. The performance estimates described subsequently are based on the use of dry air, dry fuel (water vapor may be present but no liquid water), and 100% combustion efficiency, but all other boiler losses were calculated using the detailed boiler heat balance procedure described in Ref. 124.

A more direct method was used to calculate the performance of waste-heat recovery COGAS systems. For these systems, the gas turbine performance was estimated using a simplified version, described below, of the detailed performance procedure described in Part Two, Section II on base-load gas turbines. These gas turbine performance estimates included the effects of increased back pressure on the gas turbine caused by the pressure drop through the steam boiler. The steam system performance was estimated by first calculating a detailed boiler heat balance to determine the steam conditions and the amount of steam which could be generated from the gas turbine exhaust gases. These detailed heat balances included computation of the heat transfer duties of each section of the boiler and estimates of the gas, steam, and feedwater temperatures throughout the boiler to ensure that the minimum pinch temperatures (50 F and 100 F for gas-feedwater and gas-steam temperature differences, respectively) were not exceeded. The steam turbine output was then calculated based on representative expansion line data for steam turbines.

Since many steps in the computational procedures described above require numerical iteration to achieve reasonable accuracy, a high-speed computer program was written to facilitate computation of thermodynamic efficiency, output power, fuel consumption, engine air flow, and other useful thermodynamic quantities for COGAS systems. This computer program is very flexible in that it can calculate the performance of all five basic types of COGAS systems described previously. The program can be used to calculate the degradation in performance of a standard steam cycle due to the reduced steam extraction for feedwater heating, or the performance of a steam cycle specifically designed for COGAS applications can be inputed. In addition, the program can also calculate the detailed boiler heat balances required for waste-heat recovery COGAS systems.

In order to speed calculations for many of the COGAS configurations which were of interest, as well as to enhance the flexibility of the COGAS performance program, several approximate gas turbine performance relationships were incorporated into the program. For example, since the program does not calculate detailed design parameters for the gas turbine (such as number of stages, stage geometry, etc.) it is not possible to estimate the amount of turbine cooling flow which would be necessary to cool the hot engine parts. Consequently, calculated results from the detailed gas turbine design program described in Part Two, Section II were used to develop analytical expressions for cooling flow in terms of hot gas temperature, maximum allowable metal temperature, and turbine cooling air temperature. Using a similar procedure, an analytical expression was developed to assess the gas turbine performance penalty associated with the cooling air flow. Even with these simplifications, however, the program results are sufficiently accurate for study purposes (generally within 3% or less).

Selection of Waste-Heat Recovery COGAS System

Preliminary performance calculations for the five basic types of COGAS systems are presented in Fig. 67. These calculations were made early in the contract performance period before optimum gas turbine cycle conditions were identified and before the characteristics of desulfurized fuels were known. Therefore, the operating conditions specified in Fig. 67 were used solely for the purpose of comparing the performance of the five COGAS systems and do not reflect conditions anticipated in future COGAS systems.

The calculation results presented in Fig. 67 clearly indicate that the performance of the gas-generator and two-pressure supercharged COGAS systems are so low relative to the other system types that they can be eliminated immediately from further consideration. The thermal efficiency of exhaust-fired COGAS systems would vary with power system output per unit airflow as shown in the figure. The power per unit airflow could be increased (i.e., move from left to right along curve) by increasing the amount of supplementary firing in the boiler and increasing the steam turbine output. Since all supplementary fuel fired in the boiler would be converted into useful power at the steam system only, the resultant COGAS system efficiency would tend to decrease with increased supplementary firing and increased steam turbine output. This trend of decreased efficiency with increased steam turbine output is also illustrated in Fig. 68 where power system efficiency is plotted against the gas turbine output, expressed as a percentage of total power system output. The efficiency estimates in Fig. 68 for 2400 F turbine inlet temperature are somewhat less than those shown in Fig. 67 because a much more conservative value of steam cycle efficiency, 34%, was assumed for Fig. 68. As shown in Fig. 68, the trend of increased efficiency with increased gas turbine participation exists for all exhaust-fired systems with advanced-cycle gas turbines. The efficiency trends depicted in Figs. 67 and 68 are consistent with other study results reported in Refs. 125 and 126.

According to capital cost estimates presented in a subsequent discussion of economics, the specific cost for steam systems, approximately $140/kw to $180/kw, would be higher than the specific cost for gas turbines, approximately $60/kw to $80/kw. Therefore, the specific cost of exhaust-fired systems should increase with increased steam turbine output. Consequently, those exhaust-fired systems with little supplementary firing (high ratio of gas turbine to steam turbine output) should have both lowest cost and highest performance leading to lowest busbar cost of electricity.

A similar situation would exist for supercharged systems, i.e., systems with a high gas turbine participation (corresponding to low values of power per unit airflow) should have highest performance and lowest cost leading to the lowest busbar cost of electricity. However, at very low values of power per unit airflow, supercharged systems are not much different from exhaust-fired systems, and in the limit, both systems degenerate to the waste-heat recovery type of COGAS system as illustrated in Fig. 67. There may be a limited region, depicted by the dashed lines in Fig. 67, in which a small amount of supplementary firing in both exhaust-fired and supercharged systems might provide a slight increase in efficiency. Detailed analysis of this region is beyond the capability of the computer program developed during the contract.

It should be noted that the turbine inlet temperatures assumed in Figs. 67 and 68 (above 2000 F) represent values considerably higher than the 1700 F to 1800 F temperature range utilized in existing base-load gas turbine engines. If a COGAS system were to be selected based on these low turbine inlet temperatures, a good case could be made for selecting the supercharged system depicted in Fig. 62. At low turbine inlet temperature, the efficiencies of these COGAS systems would not increase significantly with increased gas turbine participation. Instead, over a wide range of gas turbine participation the efficiencies of both systems would be between approximately 37% and 40%, with the efficiency of the supercharged system being slightly higher (approximately 1/2 point in efficiency) than that for the exhaust-fired system.

Perhaps a more significant advantage of the supercharged system in the low-temperature case would be the possibility that supercharged systems might ultimately be capable of burning coal directly. The prospects of utilizing high-heat-release, fluidized-bed combustors make supercharged systems even more interesting because then the power system efficiency would not have to be debited for energy losses from the fuel conversion system. However, several problem areas still exist with pressurized fluid-bed combustors, such as high bed pressure losses and excessive solid carry over from the fluid bed. Furthermore, it is doubtful that fluid-bed sulfur absorption schemes could achieve the high sulfur removal rates anticipated for future power stations (see Part One). In addition, fluid-bed combustors are limited to bed temperatures of 1800 F or less, thus preventing subsequent utilization of advanced high-temperature gas turbines.

One supercharged system is currently being erected in West Germany (Ref. 127). This system will employ coal gasification instead of direct coal combustion, and the turbine inlet temperature will be approximately 1500 F. This type of plant is economical in Europe, but not in the United States, for two reasons. First, fuel costs in Europe are much higher than in the United States so that a higher premium is placed on small differences in efficiency. Second, US boiler manufacturers are unwilling to accept the high development costs associated with supercharged boilers. In any case, as turbine inlet gas temperatures are increased to 2000 F and above, the waste-heat type of system will be capable of the same levels of efficiency as supercharged systems, but with lower capital costs because of the higher gas turbine participation.

Based on the previous discussions, considerations of both economic factors and future growth potential tend to favor the waste-heat recovery type of COGAS system over all other types. This system appears to offer the greatest potential for very high performance, low cost, and simple design. Consequently, all performance, economic, and design discussions of COGAS systems in subsequent sections of this report are based on waste-heat recovery systems.

Waste-Heat Recovery COGAS System Performance Trends

Preliminary performance estimates and general performance trends for waste-heat recovery COGAS power systems are presented in the following paragraphs. These performance estimates illustrate the significance of supplying high-pressure, high-temperature, low-Btu fuel gas to the gas turbine and of varying gas turbine cycle parameters on power system efficiency. In addition, brief discussions are presented of part-load operation and the use of compound-cycle gas turbines in waste-heat systems.

Parametric Performance Results

Preliminary parametric performance plots for waste-heat recovery COGAS systems are presented in Fig. 69 for turbine inlet temperatures of 2000 F, 2400 F, and 2800 F and for a range of applicable pressure ratios. Performance estimates are presented for systems burning methane and a low-Btu fuel, Fuel Gas I, supplied at burner pressure and 150 F. Both system efficiency and power output per unit airflow would be greater for systems burning the low-Btu fuel gas than for systems burning methane because this fuel gas would displace some of the air which would normally be compressed in the compressor section of the gas turbine. As a result, there would be less compressor work, higher gas turbine output, and higher power system efficiency. The Fig. 69 performance results are based on the use of compressor discharge air for cooling the hot turbine sections. Based on this cooling technique, the three turbine inlet temperatures presented in Fig. 69 would be representative of turbine inlet temperatures projected to be available for the three generations of power systems. Using this cooling technique, net system efficiencies are projected to increase from 45% for first generation systems to 55% for third generation systems.

By utilizing advanced cooling techniques such as precooling of the turbine cooling air, the maximum turbine inlet temperatures for the three generations are projected to be 2200 F, 2800 F, and 3100 F, respectively, and would result in efficiencies several points higher than those depicted in Fig. 69. The projected efficiencies of COGAS systems burning Fuel Gas I and using cooling air precooled to 125 F are shown in Fig. 70 to approach 47%, 55%, and 58%, respectively, for the three generation power systems. This figure also illustrates the large increase in power system efficiency that results from recovering gas turbine waste heat. The magnitude of this waste heat is represented by the gas turbine exhaust temperatures plotted in Fig. 71.

All performance data presented in Figs. 69, 70, and 71 are based on low gas turbine back pressure corresponding to relative low gas-side pressure drop (8.3 in. H_2O) through the waste-heat boiler. The variation in COGAS system efficiency with boiler pressure drop is relatively slight, as illustrated in Fig. 72. These results indicate that it should be possible to design forced-convection boilers with relatively high pressure drop (between 20 and 100 in. H_2O) leading to compact, low-cost installations. Waste-heat boiler design analyses are discussed in the ensuing section on design considerations.

The power system fuel flow rates calculated on the basis of the system efficiencies presented in Fig. 70 could be reduced still further if the fuel gas were supplied to the power system at elevated temperature. The percent reduction in power system fuel flow rate is plotted against fuel supply temperature in Fig. 73 for conditions representative of third-generation systems and a base supply temperature of 232 F. If power system efficiency were defined on the basis of heating value of the fuel supplied to the gas turbine, instead of heating value plus sensible heat as defined in the preceding efficiency definitions, then power system efficiency would increase by the same percentage that fuel flow rate would decrease. By using the definition given previously, however, power system efficiency would not change significantly with increased fuel temperature. Since the net cost of fuel from a high-temperature fuel supply system was reported in Part One to be higher than the cost of fuel from low-temperature supply systems, subsequent design studies were based on relatively low-temperature (232 F) fuel.

Part-Load Performance

Since the thermal efficiency of fuel conversion processes discussed in Part One would not change significantly at reduced output (at least down to 50% of rated output), part-load characteristics of integrated power stations would be governed by the power system. As indicated in the discussion of part-load characteristics of gas turbines in Section II, analytical determination of part-load performance of gas turbines would require the use of relatively complex engine matching procedures and involve a detailed knowledge of the off-design performance of each gas turbine component. Unless representative off-design performance maps for the compressor and turbine components are available, the off-design performance of these components must be estimated by analyzing the individual velocity triangle diagrams of every compressor and turbine stage. Estimation of off-design characteristics of the steam power system components would also involve the use of complex part-load procedures. Such detailed analyses are beyond the scope of the present study. In any case, part-load operation of waste-heat COGAS power systems would be of secondary importance because, with their very high projected operating efficiency it would be desirable to operate them at maximum output. Furthermore, the high power station load factor, 70%, selected for base-load performance comparisons would require that these power systems operate at or near rated output most of the time. Thus, because of its relative unimportance and because of the complexities involved, the part-load characteristics of COGAS system were not investigated in detail during this study.

Nevertheless, the part-load performance characteristics of advanced waste-heat recovery COGAS systems considered herein should exhibit the same general part-load performance trends exhibited by representative present-day waste-heat systems. Part-load power in present-day waste-heat systems is usually achieved by reducing the turbine inlet temperature of the gas turbine. In this manner, the gas turbine output power is reduced significantly. There is also a slight decrease in boiler gas flow rate which contributes to the decrease in steam turbine power output. Part-load performance of representative present-day systems, based on data from Refs. 125 and 128, is illustrated in Fig. 74. The two COGAS systems represented in this figure utilize two-pressure steam systems in order to increase power output from the steam system and achieve high system efficiency, whereas the advanced COGAS systems described herein would be capable of achieving high efficiency with single-pressure steam systems. If this method of part-load operation were used in future waste-heat systems, no modifications to the gas turbine designs presented herein would be required.

If higher part-load efficiency than that indicated in Fig. 74 were desired for advanced COGAS systems, the gas turbine could be equipped with variable-geometry guide vanes before the power turbine in order to reduce engine gas flow rate. This method of operation at part-load power would allow the gas turbine to operate at the full-load turbine inlet temperature leading to improved gas turbine part-load efficiency. In addition, the high temperature of the gases exhausting into the steam boiler would be maintained, thus permitting generation of the same high quality, although smaller quantity, steam as that generated at full load. It should be noted that variable geometry guide vanes, which would add to the complexity and cost of the gas turbines, have not bee included in the gas turbine designs considered herein.

COGAS Systems with Compound-Cycle Gas Turbines

Preliminary performance investigations were made of waste-heat recovery COGAS systems utilizing gas turbine cycles incorporating intercooling, reheat, regeneration, and all combinations thereof. In general, regeneration was found to be incompatible with steam waste-heat recovery because both of these recovery schemes compete for low-grade heat rejected from the gas turbine. Insertion of a regenerator into a COGAS system would require that the steam system operate with reduced hot gas supply temperature, reduced steam conditions and, therefore, reduced steam cycle performance. Although heat recovered in a regenerator would be converted into power at a higher efficiency than heat recovered in the steam cycle, the degradation in steam cycle performance would result in an overall reduction in COGAS system performance. This result was observed for all COGAS systems utilizing regeneration, either alone or in combination with reheat and/or intercooling.

The projected performance of second-generation COGAS systems incorporating reheat between the compressor-drive and power turbines is shown in Fig. 75. The use of reheat in this type of system might lead to increases in both efficiency and specific output. However, the use of a low-Btu fuel might not permit the use of reheat since the amount of oxygen in the gases exiting from the compressor-drive turbine appear to be insufficient to support combustion in a reheater. Since study of flammability limits of low-Btu fuel gas in vitiated atmospheres was beyond the scope of the present investigation, analysis of reheat cycles was not pursued.

Since compressor intercooling, by itself or in combination with reheat, does not result in a significant improvement in efficiency (see Fig. 75), intercooling was not considered further. As a result of these preliminary studies, all subsequent analyses of COGAS systems were based on the use of simple-cycle gas turbines.

WASTE-HEAT RECOVERY COGAS SYSTEM DESIGN CONSIDERATIONS

The COGAS power system having the greatest potential for future high-performance and low-cost power generation was previously identified to consist of simple-cycle gas turbines followed by a waste-heat recovery steam system. In the following paragraphs, simple-cycle gas turbine design and cost considerations are combined with performance predictions from the previous section in order to select the gas turbine and steam system design parameters which lead to estimates of busbar power costs.

Gas Turbine Design Parameters

The primary gas turbine design parameters such as turbine inlet temperature, compressor pressure ratio, and gas generator configuration (single or twin-spool) were selected by considering variations in capital cost of the engine and variations in fuel cost as reflected by variations in cycle efficiency.

In COGAS systems it is always desirable to use the maximum turbine inlet temperature that gas turbine technology will allow. This is true because gas turbine performance generally improves with increasing gas temperature, whereas gas turbine cost per unit of output generally decreases with increasing temperature. The specific relationships between gas turbine costs, performance, and gas temperature are discussed more fully in Part Two, Section II. In addition to these variations in gas turbine performance and cost, higher turbine inlet temperature would result in higher turbine exhaust temperature (for fixed pressure ratio) which would enable more power to be extracted from the waste-heat steam system. Higher gas temperature into the steam boiler would also lead to lower boiler cost per unit of heat transfer surface area because of the larger gas-steam temperature difference. Therefore, the highest gas turbine inlet temperature possible for each generation should be chosen. The gas turbine analyses presented in Part Two, Section II indicate that turbine inlet temperatures of 2200 F, 2800 F, and 3100 F should be technologically feasible for first-, second-, and third-generation systems, respectively, provided the turbine cooling air can be precooled to approximately 200 F and the historic trend in materials improvements continues. Consequently, these temperatures were assumed for the three generations of power systems described herein.

Identification of the optimum pressure ratio for each generation of COGAS system is more involved than identification of inlet temperature and requires thorough investigation of performance and cost trends. The preliminary performance estimates presented in Fig. 70 for Fuel Gas I were used in the selection of pressure ratio. In gas turbine-type power systems, it is desirable to operate both at high power system efficiency, in order to reduce fuel cost, and at high values of power output per unit airflow, in order to reduce the cost of air and gas handling equipment. Thus, it appears from Fig. 70 results that a pressure ratio of 8:1 would be most desirable for first-generation systems. An engine with this low pressure ratio would have only one compressor spool.

Based on the Fig. 70 results, the optimum pressure ratio for second-generation systems would be between approximately 12 and 16. Preliminary cost estimates for single-spool gas turbines derived using the gas turbine costing procedure described in Appendix E indicated that an engine with a pressure ratio of 16 would cost approximately $2.00/kw more than an engine with a pressure ratio of 12. A $2.00/kw additional expenditure could not, however, be justified on the basis of the slight efficiency differential between systems with these two pressure ratios. Furthermore, startup problems with a 16:1 pressure ratio single-spool engine could be serious, and could necessitate the use of a twin-spool design. The cost of a twin-spool engine with a pressure ratio of 16 was estimated to be approximately $1/kw more than the 12:1 single-spool engine. In this case, it was estimated that the higher fixed charges for the higher pressure ratio system would be approximately offset by reduced fuel costs. However, twin-spool engines are less desirable than single-spool engines because twin-spool engines require more bearings, and from a reliability standpoint, it would be desirable to minimize bearings. Also, the 12:1 pressure ratio gas turbine would have a higher gas turbine exit temperature leading to a lower-cost steam boiler. Consequently, the 12:1 single-spool engine was selected for second-generation power systems.

For third-generation power systems, the optimum pressure ratio would be between 16 and 24. For these high pressure ratios, twin-spool designs would be required. Preliminary cost estimates were prepared for gas turbines with pressure ratios of 20 and 24. The 20:1 pressure ratio engine was found to cost approximately $0.5/kw less than the 24:1 pressure ratio engine. Again, the increased capital cost and reduced fuel costs would be approximately compensating. Since the lower pressure ratio engine would result in a more conservative engine design and yield a higher gas temperature for the steam boiler leading to lower boiler cost, the 20:1 pressure ratio twin-spool engine was selected for third-generation power systems.

Waste-Heat Steam Cycle Analyses

Extensive performance, design, and economic analyses were conducted in order to select the waste-heat steam cycle which would result in highest station performance and lowest-cost electricity. Both single-pressure and dual-pressure steam systems were considered in order to recover maximum gas turbine exhaust heat and achieve low stack gas temperature. In the following paragraphs, main steam refers to the high-pressure steam system and secondary steam refers to the low-pressure steam system (whenever employed).

For all cycles examined,the steam turbines were assumed to be of the condensing type with 2.0 in. Hg exhaust pressure. Since the feedwater would be heated in the economizer using low-grade heat from the stack gases, no extractions from the steam turbine would be necessary for feedwater heating. Also, instead of using a deaerating feedwater heater, a deaerating condenser followed by chemical scavenging of oxygen was considered to provide adequate feedwater treatment. Minimum pinch temperature differences of 50 F and 100 F were assumed for gas-to-liquid and gas-to-steam heat transfer in the boiler, respectively.

Main Steam Systems

A wide range of steam conditions was considered for the high-pressure or main steam cycles as indicated in Table 70. Steam cycles with very low steam conditions, in general, would result in lower power system performance than cycles with high-quality steam and were not considered in detail. Table 71 gives the results of performance comparisons for steam systems with reasonably high-quality inlet steam. Several simplifying approximations were made for these calculation results as indicated in Table 71. The significant waste-heat steam system performance parameter presented in the table is the output power per unit of gas flow through the boiler. Note that the system with highest efficiency (lowest heat rate) would not lead to maximum output. The maximum differential in output per unit of gas flow between steam systems in Table 71 is only 5.3% (2400 psig/1000 F/1000 F cycle vs 3500 psig/1000 F/770 F cycle). The supercritical cycles (3500 psig) definitely would be inferior to the subcritical cycles for these waste-heat application because in addition to having poorer performance, they would be more expensive. The estimated performance differentials between the subcritical 2400 psig and 1800 psig cycles, howerver, are very small.

The subcritical cycles that appear to be most promising are the 2400 psig/1000 F/1000 F and the 1800 psig/1000 F cycles. A preliminary economic comparison between steam systems employing these two cycles was made based on estimated costs for the waste-heat boiler and the steam turbine generator set only. The costs of other equipment were assumed to be the same for both systems. This appears to be a good approximation because, while the 2400 psig system would require additional capital for high-pressure main steam piping and reheat piping, the 1800 psig system would have a 23% higher flow rate requiring larger-diameter steam piping and a larger condenser. This economic comparison was based on a gas turbine exhaust temperature of 1226 F and a gas flow rate of 3600 lb/sec which would yield a nominal COGAS system output of 1000 Mw. The net output from the 2400 psig and the 1800 psig steam systems would be approximately 342 Mw and 337 Mw, respectively. The 2400 psig system would have 5 Mw of additional output which, at 6 mills/kwhr, would equate to an additional capital investment of $1,290,000. Furthermore, the 2400 psig system was estimated to cost approximately $1,500,000 less than the 1800 psig system. Thus, a total cost advantage of approximately $2,790,000 exists in favor of the 2400 psig system.

Based on this preliminary analysis, the 2400 psig/1000 F/1000 F cycle with higher performance and lower cost than other waste-heat cycles was the obvious choice for use in all waste-heat COGAS systems in which the gas temperature entering the boiler would be high enough to raise 1000 F steam, i.e., greater than approximately 1100 F. Since this condition would be met for all second- and third-generation COGAS systems of interest (turbine inlet temperature above 2800 F) and for all first-generation systems (2200 F turbine inlet temperature) with gas turbine compressor pressure ratios less than 13 (see Fig. 71), main steam cycles with reduced steam conditions were not examined further.

Having selected the 2400 psig/1000 F/1000 F steam cycle, its performance characteristics were investigated in more detail. A detailed heat balance for this steam cycle is illustrated in Fig. 76. All flow rates in this figure are expressed as a percent of main (throttle) steam flow because the absolute flow rate for any specific COGAS system would depend on the specific power split between the gas turbine and the steam system. All significant leakage flows as well as intermediate temperatures, pressures, and enthalpies are identified in the figure. The condensate and boiler feed pumps would be driven by electric motors. Since there would be no extraction from the steam turbine for feedwater heating, the steam flow to the condenser would be over 99% of the main steam flow, instead of only 65% of main steam flow as in the case of conventional steam systems, and the temperature of feedwater supplied to the boiler economizer would be only 110 F. The net steam cycle efficiency is estimated to be 38.8%. The final stack temperature, amount of steam generated in the boiler, and the steam system output power are shown in Figs. 77, 79, and 79, respectively. The results shown in these three figures are representative of conditions anticipated in future COGAS systems and are relatively insensitive to fuel type and fuel/air ratio. Nevertheless, the specific fuel type and calculated fuel/air ratio were used in all detailed boiler calculations made using the computer program described previously. In those COGAS systems requiring precooling of the turbine cooling air, the 110 F feedwater would be piped to the precooler and used to lower the cooling air temperature to approximately 200 F. The net steam cycle thermal efficiency would be approximately 39% to 40%, depending on the amount of precooling required, and the final stack temperature would be slightly higher than the Fig. 77 results.

Secondary Steam Systems

The data in Fig. 77 indicate that for high gas temperature into the boiler and 110 F feedwater temperature, stack temperatures in the 200 F to 300 F range would be possible. In some compound-cycle COGAS systems which were briefly examined, the boiler feedwater could be utilized in intercoolers to cool the compressor air and reduce compressor power. It was found that in some of these cycles, the feedwater could be heated from 110 F to over 400 F in the intercooler, depending on the compressor pressure ratio and feedwater flow rate. As a result of this higher final feedwater temperature, the final stack temperature would increase to somewhat over 400 F. In the event that these compound cycles with compressor intercooling might look attractive, several low-pressure secondary steam cycles were considered for recovery of very low-grade heat rejected from the main steam cycle. However, as discussed in a previous section on system performance trends, simple-cycle gas turbines gave higher performance than intercooled cycles so that consideration of secondary steam systems was subsequently discontinued. The following analysis of secondary steam systems is presented solely of the sake of completeness.

Both saturated and superheated throttle temperatures were evaluated for secondary steam systems while varying throttle pressure from 3 to 80 psia. A constant steam turbine efficiency of 83.5% was assumed. For fixed gas temperature into the secondary boiler, a range of throttle steam conditions was considered, and the conditions which yielded maximum power output were selected. This procedure was followed for a range of gas temperatures yielding the results shown in Fig. 80. Typical optimum steam conditions are depicted in this figure for several gas temperatures.

Since the output power from the secondary steam system would be much less than the output power from the main steam system, preliminary cost estimates for secondary steam systems were prepared to evaluate their economic feasibility. For the purposes of these preliminary cost estimates, the gas flow rate was assumed to be 3600 lb/sec, and the gas temperature into the secondary boiler was assumed to be 366 F which corresponds to the stack temperature for the 2400 psig/1000 F/1000 F main steam system shown in Table 71. The optimum throttle steam conditions for this system were found to be 14.7 psia and 266 F, and the output power was found to be 4.28 kw/lb gas/sec or approximately 15,400 kw. The final power system stack temperature would be 250 F.

Three different configurations for the secondary steam system were considered for costing purposes as follows:

Case I - Secondary steam system completely separate from main steam system, i.e., separate steam turbine, electric generator, secondary waste-heat boiler, piping, feed pump controls, steam condenser, etc.

Case II - Separate boilers, piping, feed pump, and controls but exhaust end of main steam turbine enlarged to accommodate admission of secondary steam. Main steam condenser and electric generator also enlarged.

Case III - Separate boiler, piping, feed pump, controls and low-pressure steam turbine. Low-pressure turbine coupled to shaft of main steam turbine. Exhaust from low-pressure turbine sent to enlarged main steam condenser. Main electric generator also enlarged.

Prices for these special low-pressure steam turbines and enlarged main turbines were obtained from turbine manufacturers, while the costs for condensers were estimated based on recent quotations for similar components. Costs for the secondary system waste-heat boilers were estimated using heat transfer surface cost correlations. These estimated costs, in $/kw, are summarized in Table 72.

The subtotals for the turbines, condensers, boilers, and incremental building costs (over that required for the main turbine building) ranged from \$162/kw to \$189/kw. The remaining plant construction costs, consisting primarily of foundations, structures, piping, controls, and installation, are estimated to cost \$100/kw resulting in total installed costs of \$262/kw to \$289/kw. Case II was eliminated from consideration because one turbine manufacturer indicated that the problem of controlling two boilers separately and feeding the output from each boiler through a single steam turbine would lead to very difficult control problems. The Case III configuration, which would cost approximately 10% less than the Case I configuration, would be chosen.

If the estimated capital cost for the secondary system of \$262/kw is amortized at 14% annual charges and 70% annual load factor, a generation cost of approximately 6 mills/kwhr would result, neglecting miscellaneous labor and operating expenses. No fuel costs are charged to the secondary system because the steam would be generated using waste heat. Since 6 mills/kwhr is also the approximate present-day cost for generating electricity, the secondary steam system analyzed herein is regarded as being economically marginal. If the gas temperature into the secondary boiler were less than 366 F, the power output per unit gas flow would decrease (see Fig. 80), the specific cost would increase, and secondary systems would not be economically feasible. Conversely, if the secondary system inlet gas temperature were 400 F or above, the power output per unit gas flow would increase, the specific cost would decrease, and the use of secondary steam systems would be economically feasible.

Waste-Heat Boiler Design Analysis

The waste-heat boiler compromises a significant portion of the heat recovery steam system capital cost. One method of reducing the cost of these boilers would be to design them for relatively high gas-side pressure drop. High pressure drop, however, would have an adverse effect on power system efficiency, as depicted in Fig. 72, resutling in increased fuel costs. Consequently, a detailed design and economic trade-off analysis was carried out to determine the gas-side pressure drop in the boiler which would result in lower power cost. A separate analysis was conducted to determine the optimum boiler tube diameter. In addition to cost considerations, maintenance, operating, and structural problems associated with high-pressure boiler enclosures influenced selection of boiler pressure drop and tube diameter.

All boilers were designed to generate throttle and reheat steam for the 2400 psig/1000 F/1000 F steam cycle depicted in Fig. 76, starting with 110 F feedwater. Several boiler manufacturers indicated that they believe the 110 F feedwater temperature to be too low and might lead to sulfidation corrosion on the gas side. Since the boiler gases would be derived by combustion of very-low-sulfur fuel, low-temperature sulfidation was not considered to be a problem. Additional concern was expressed over the possibility of achieving sufficient deaeration of the feedwater in the deaerating condenser. It appears, however, that sufficient deaeration could be achieved with a deaerating condenser followed by chemical scavenging of the oxygen. In any case, the boiler design could be modified to provide for high-temperature deaeration and more complete oxygen removal by splitting the economizer into low- and high-pressure sections separated by a deaerating drum. The cost of these modifications should be nominal and were not considered herein.

Boiler Design Analysis

The heat recovery boiler design is based on forced circulation of the hot gases which allows for very compact arrangement of the heating surfaces. The heat transfer surfaces are arranged such that the hot gases would pass in crossflow relative to the water and steam flows. The heating surfaces comprise the following sectons: economizer, vaporizer, primary (or low-temperature) superheater, reheater, and secondary (or high-temperature) superheater. The superheater surface was arbitrarily divided into two sections to alleviate the gas-steam pinch problem that would occur whenever the gas temperature entering the boiler would be less than approximately 1200 F.

The boiler designs were based on procedures and design data contained in Refs. 129, 130, and 131. Sizing of the boiler was determined by iterative calculations for each of the five boiler sections for various gas velocities selected to result in the total allowable boiler pressure drop. The required gas velocity for each section was obtained by variation of the number of tubes arranged across the gas flow path.

The analysis of boiler cost variation with pressure drop was based on the use of 2.375 in. OD pipe for all heat transfer sections. Finned seamless carbon steel schedule 160 pipe was assumed for the economizer. The carbon steel fins would be 0.035 in. thick and 0.250 in. high, spaced 10 fins per inch of pipe length. The primary superheater pipe was assumed to be schedule 160 seamless grade P-11 alloy steel (1¼% Cr-½% Mo). The secondary superheater pipe was assumed to be XX strong seamless grade P-22 alloy steel (2¼% Cr-1% Mo). The reheater was also assumed to be P-22 alloy steel but schedule 80 pipe. The pipe length for all sections was assumed to be 30 ft, and the spacing between pipe centers was assumed to be 5 in. to allow adequate gas flow pasage area.

A subsequent analysis was conducted to compare the use of boiler tubes with boiler pipe. Tubes are more desirable than pipes because tubes are available in a wider range of tube wall thicknesses, thus leading to lighter-weight and more compact designs. Tube diameter was varied from 1.625 in. OD to 2.750 in. OD. The tube lengths were also assumed to be 30 ft, but the ratio of tube spacing between centers to tube OD was maintained at a constant value of 2.1. Tube materials were taken to be the same as the pipe materials for corresponding boiler sections.

Boiler Cost Analysis

The boiler manufacturers were generally unresponsive to requests for boiler cost information. Therefore, it was necessary to use past experience and engineering judgment to establish a procedure for estimating the cost of waste-heat boilers based on the total heat transfer surface area required, the cost of the enclosure (setting, insulation, supporting structural steel, and lagging), and the erection cost. A single cost estimate received from one boiler manufacturer at the end of the study period for one representative waste-heat boiler design was approximately 5.5% less than the corresponding estimate derived using the costing procedure. This relative agreement was considered to lend credibility to the boiler cost estimates presented herein.

The cost of the heat transfer surface was based on the type of material and the total surface area required in each section of the boiler. The costs per unit of heat transfer surface area ssumed for each section of the boiler are shown in Table 73.

The cost of the boiler enclosure was assumed to be proportional to the heating surface and proportional to the gas pressure in the boiler. Enclosure costs were calculated by the following equation:

$$E = H(0.15 + 0.002\ P)$$

where E = cost of boiler enclosure, \$
H = cost of all heating surfaces, \$
P = inlet gas pressure to boiler, in. H_2O

The boiler erection cost was assumed to be 30% of the delivered equipment cost. All controls, foundations, contingency, undistributed costs, and escalation charges were accounted for by adding 24% to the calculated erected boiler cost. The resulting boiler cost subtotal was used in the comparative boiler design analyses. An additional 15%, however, was added to the final power system boiler cost estimates to account for engineering, overhead, and profit for the boiler manufacturer.

Results of Boiler Analyses

The boiler design and economic analyses are based on preliminary COGAS system performance and boiler heat balance calculations made using the COGAS computer program described previously for conditions representative of second-generation power systems burning Fuel Gas I. These preliminary performance and heat balance calculations are presented in Table 74 for five different gas turbine back pressures ranging from 8.3 to 200 in. H_2O gauge. For the purposes of this comparative analysis, the boiler gas inlet pressure was assumed to equal the gas turbine back pressure. In actual practice, there would be additional pressure losses in the breeching and silencing equipment, although some of this pressure loss would be compensated for by the draft created by the hot gases existing the stack. This additional back pressure should not exceed 10 in. H_2O.

The heat transfer surface areas for each boiler section and the total boiler heat transfer surface area, corresponding to the data in Table 74 are given in the upper portion of Table 75. Total boiler heat transfer area is also plotted against gas turbine back pressure in Fig. 81. These data clearly show that the required surface area diminishes as gas pressure is increased, although the effect of gas pressure is less pronounced above 100 in. H_2O. Below 20 in. H_2O, the required surface area increases sharply.

Also shown in Table 75 are estimated costs for the steam boiler with 2.375 in. OD pipe, power outputs from the gas turbine and steam system, and differential outputs relative to the base case with 50 in. H_2O gauge back pressure. The differential (or incremental) cost for the gas turbine and steam system were taken to be \$30/kw and \$65/kw, respectively, leading to the total differential gas and steam turbine generator plant costs tabulated. By combining differential boiler and turbine generator plant costs the total power system differential capital costs were obtained. Using the assumed fixed charge rate of 14\$, the differential annual fixed costs presented in Fig. 82 were obtained. Also shown in this figure are differential annual fuel costs, based on an average equivalent fuel cost of \$0.40/10^6 Btu delivered to the gas turbine and operation for 6200 hr/yr which corresponds to an average annual load factor of approximately 70%.

The total evaluated differential annual cost estimates are presented in Fig. 83. As shown in this figure, the lowest cost would be obtained for back pressures in the range of 40 to 50 in. H_2O gauge. Since a lower back pressure would lead to less boiler gas leakage and fewer casing and insulation problems, the back pressure of 40 in. H_2O was selected for second-generation boiler designs. It should be noted, however, that for pressures as low as 20 in. H_2O and as high as 80 in. H_2O, the differential annual costs would be approximately \$120,000/yr which corresponds to approximately 0.02 mills/kwhr. Thus, boiler pressure drop is not a critical design parameter and has only a second-order effect on electric power cost.

The above results were based on preliminary second-generation power system designs. Similar results were obtained for preliminary first- and third-generation power systems. Because of the relative insensitivity of electric power cost to back pressure, 40 in. H_2O back pressure was selected for COGAS systems of all three time generations.

The results of the analysis to determine the effect of using tubes in the boiler instead of pipes are summarized in Table 76. From these results it may be seen that boilers fabricated from tubes would cost substantially less than boilers fabricated from the 2.375 in. OD pipe. Furthermore, the 2.000 in. OD tube appears to yield the lowest boiler cost, although the cost differential between the 2.000 and 2.375 in. OD tubes is very small. Since the 2.375 in. diameter tubes would be stiffer than the 2.000 in. tubes leading to greater mechanical and structural integrity for the boiler, the 2.375 in. diameter tubes were selected for the final waste-heat boiler designs.

CHARACTERISTICS OF SELECTED COGAS POWER SYSTEMS

The design, performance, and economic characteristics of the waste-heat recovery COGAS power systems selected for each of the three generations are presented in the following paragraphs. These future power systems would consist of high-temperature, simple-cycle gas turbines exhausting into compact, forced-convection waste-heat recovery boilers which generate steam for nonregenerative steam systems. All power system designs are for 1000-Mw net output power from the power system and are based on the use of low-Btu fuel gas supplied to the gas turbine at high pressure and approximately 230 F temperature.

Design Characteristics

Specific design characteristics for the gas turbine portions of the selected COGAS power systems are summarized in Table 77. The first- and second-generation designs incorporate single-spool gas generators, whereas the third-generation design incorporates a twin-spool gas generator. All three gas turbines would have a separate power turbine rotating at 1800 rpm for direct drive of the electric generator. The gas turbine inlet temperatures were taken to be 2200 F, 2800 F, and 3100 F, respectively, and the compressor pressure ratios were taken to be 8, 12, and 20, respectively, for the three power systems. The overall gas turbine lengths (from compressor inlet stage to power turbine exit stage) are shown in Table 77 to be approximately 33, 27, and 26 ft, respectively, for the three generations.

Selected design data for the steam portions of the COGAS power systems are summarized in Table 78. For each design, the steam turbine would be of tandem-compound, double-flow construction. The throttle steam conditions would be 2400 psig and 1000 F. There would be a single reheat to 1000 F, but there would be no steam extraction for feedwater heating. The forced-circulation, waste-heat boilers would be designed for a total gas-side pressure drop of 40 in. H_2O. The gas, steam, and feedwater temperatures at selected points in the boiler are shown in the table along with calculated heat transfer duties and surface areas for each section of the boiler.

Performance Characteristics

Selected performance data presented in Table 79 indicate the relative gas turbine output, net steam system output, and power system auxiliary requirements. Also shown in the table are thermal efficiencies for the gas turbines and steam cycles. The net power system efficiencies are shown in the table to be 47.0%, 54.5%, and 57.7% for the three generations. These very high thermal efficiencies are one of the principal reasons why advanced cycle COGAS systems appear to be very attractive for future power generation applications.

Also shown in Table 79 are estimated values of the thermal energy which would have to be rejected to cooling water by these cycles. This thermal heat rejection would diminish from approximately 2260 Btu/kwhr for the first-generation system to approximately 1640 Btu/kwhr for the third-generation power system. Comparable heat rejection rates from fossil-fueled and nuclear steam stations are approximately 4300 Btu/kwhr and 6600 Btu/kwhr, respectively.

Economics

The detailed gas turbine and steam system design data presented in Tables 77 and 78 form the basis of power system cost estimates presented in Tables 80 through 88. These cost estimates are tabulated according to standard FPC accounts which include land and land rights, structures and improvements, prime movers (gas turbines), electric generators (for gas turbines), boiler plant equipment, steam turbine generator units, accessory electrical equipment, miscellaneous power plant equipment, and miscellaneous station equipment. Detailed cost tabulations for these eight FPC account categories are presented in Tables 80 through 87. Capital costs for the gas turbines were estimated using the gas turbine costing procedure described in Appendix F. All other cost estimates for equipment are based on detailed cost estimating procedures used to estimate costs for conventional steam power generating stations (see Appendix D). A summary of station cost estimates is presented in Table 88 for the three generations of power systems. As shown in this table, the total direct and indirect construction expenses are estimated to be approximately \$109 x 10^6, \$94 x 10^6, and \$89 x 10^6, respectively, for the three generations of power systems. Approximately 45 to 50% of these total costs are assignable to the gas turbine portion of the power system, with the balance being assignable to the steam portion.

Operating and maintenance costs for the steam portions of the power systems were assumed to be 0.2 mills/kwhr of net steam system output. The corresponding costs for the gas turbine portions of the power systems were assumed to be 0.5 mills/kwhr of net gas turbine output. Supplies and materials were estimated to cost approximately 0.2 mills/kwhr of net power system output.

The estimated electric power generation costs for the three COGAS systems are presented in Fig. 84 in terms of the total cost of fuel delivered to the power system. According to the data presented in Part One, the total cost of fuel delivered to first-, second-, and third-generation COGAS power systems from fuel conversion processes incorporating partial oxidation of coal are estimated to be approximately 42, 33, and 34¢/million Btu, respectively. Based on these fuel costs, the electric power generation costs for each power station design are estimated to be approximately 7.3, 5.3, and 5.2 mills/kwhr, respectively. The first-generation power cost estimate is approximately 1 mill/kwhr greater than the corresponding estimate for first-generation, coal-fired steam station. The power cost estimates for the second- and third-generation power stations, however, are approximately 7 to 10% less than the corresponding estimates for coal-fired steam stations. Thus, advanced-design COGAS power systems burning relatively expensive, desulfurized fuel have the potential of generating lower-cost electricity than is projected for future coal-fired steam power stations. Detailed electric power generating costs for integrated coal- and oil-fired COGAS power stations are presented in Part Three of this report.

Section IV - Topping and Bottoming Cycles

This section contains the results of technical and economic feasibility analyses of binary-fluid topping and bottoming cycles used in conjunction with conventional steam power systems. Bottoming cycles for use with gas turbine and COGAS systems were also investigated. Preliminary performance and cost estimates for these systems indicated that they could not compete favorably with conventional steam systems or advanced power systems incorporating high-temperature gas turbines.

TOPPING CYCLES

It has been known for a long time that a power cycle using a low-vapor-pressure working fluid could be operated on top of a steam cycle, thus increasing the average temperature through which heat would be added to the system, and reducing stress problems associated with the higher temperatures. During the years between 1920 and 1950, a number of power plants using mercury vapor cycles to top conventional steam cycles were built. The maximum temperature of such systems was usually limited to about 950 F. Some of these binary power plants operated quite successfully and did show increased performance over the then-current conventional steam cycles whose maximum steam temperatures were low, 500 F being not uncommon. However, these classic binary-cycle power stations have been unable to compete with more modern steam plants, which now operate at maximum temperatures of approximately 1000 F. Recently, interest in the development of topping cycles has been rekindled because of: (1) progress in liquid metal technology and accumulation of extensive liquid metal property data, (2) potential applications in nuclear power and space power systems, with the commitment of, (3) potential application for high-efficiency commercial power generation systems.

In the following paragraphs, various working fluids and topping cycle configurations are discussed. Preliminary performance and cost analyses are presented for some of the more promising topping cycles.

Working Fluid Selection

In order to achieve a high thermal efficiency for a topping cycle, substances with low vapor pressure and good corrosion properties at high temperature must be used as the working fluid. A number of organic fluids such as Dowtherm, thiophene, and the fluorocarbons decompose at temperatures above 500 F and therefore were ruled out as fluids to top a 1000 F steam cycle. Mercury is expensive ($2000/ft^3), has poisonous vapor, and has previously proved only moderately successful as a working fluid. The vapor pressures of sodium and lithium are too low at 1000 F to be suitable for the design of turbines and heat exchangers Potassium, however, displays several promising aspects; it is relatively inexpensive ($50/ft^3), it has a suitable vapor pressure in the range of 1000 F to 2000 F, and it has relatively little corrosion reaction with stainless steel up to a temperature of approximately 1600 F (Refs. 132 and 133). Potassium could also be used with coated columbium up to 2200 F (Ref. 134).

Satisfactory data for thermodynamic and transport properties of potassium are available (Ref. 135). Potassium is not toxic, although it burns in air. It also reacts rapidly with water to produce hydrogen which may create explosion hazards. Therefore, a power station that would use potassium must be designed so that the consequences of such a reaction would not be serious.

Parametric Performance Analysis

The binary power system would consist of a potassium cycle topping a steam cycle, as shown in the t-s diagram of Fig. 85. In this binary power system, heat would be added in a clean-fuel-fired potassium boiler. For a given maximum potassium temperature, it would be desirable to heat potassium at a saturation pressure corresponding to the maximum temperature, (process 2-8-2'-3 in Fig. 85), so that the average temperature of heat addition would be the highest. This heating process would imply a saturated vapor rather than a superheated vapor at the specified turbine inlet temperature. Either reheat or moisture separation at a lower pressure level would be necessary to prevent moisture erosion in the turbine. Heat rejected in the potassium condenser would be used to generate steam for the steam cycle.

A thermodynamic analysis of a binary power system indicates that the binary cycle efficiency, η_B, would be a function of the potassium cycle efficiency, η_K, and the steam cycle efficiency, η_S through the relation

$$\eta_B = \eta_K + (1 - \eta_K)\, \eta_S$$

For a steam cycle efficiency of 45% and potassium cycle efficiency of 14% to 32%, the binary cycle efficiency would range from 53% to 62%, representing an improvement of 8 to 17 points over the best steam cycle, as shown in Fig. 86.

Furthermore, thermodynamic analysis indicates that the binary cycle efficiency would be influenced by at least four characteristic temperatures, namely: potassium inlet temperature T_3, potassium condenser temperature T_1, maximum steam temperature T_s, and the steam condenser temperature T_c. Of these four characteristic temperatures, T_c would always be constrained by the cooling water temperature and has been fixed at 100 F. The choice of T_3, which depends largely on material technology, is the subject of major concern. In the present study, system performance was evaluated for six potassium inlet temperatures between 1400 and 2200 F. Even if T_3 were specified, there would still exist numerous combinations of T_1 and T_s which would give different binary cycle efficiencies.

A closer examination of the expression for binary cycle efficiency reveals that

$$\frac{d\eta_B}{d\eta_S} > \frac{d\eta_B}{d\eta_K} ,$$

indicating that an increase of η_S would be favored over an increase in η_K in order to improve the binary cycle efficiency. Based on the discussions in Section I, the highest-efficiency practical steam cycle is the 3500 psig/1000 F/1000 F cycle with a thermal efficiency of approximately 45%. With T_S set at 1000 F, the optimization of the binary cycle power systems reduces to the selection of T_1, or to ΔT, the difference between the potassium condenser temperature and the maximum steam temperature. Selection of a high ΔT would mean a smaller potassium condenser but lower η_K and η_B, whereas selection of a low ΔT would mean a larger potassium condenser although with a higher η_B. One study (Ref. 132) suggested that near-minimum cost would be obtainable for $\Delta T = 50$ F. Three potassium condenser temperatures, 1040, 1080, and 1120 F, were examined during the present study.

Three basic topping cycle configurations patterned after those reported in Refs. 135, 136, and 137 were investigated. These are: (1) simple Rankine, (2) reheat Rankine, and (3) regenerative Rankine with moisture separation. Throughout the analysis, a system pressure loss equal to 15% of the turbine inlet pressure was assumed as were a potassium turbine efficiency of 75% and a pump efficiency of 70%. The maximum potassium moisture content in any section of the turbine was limited to 12% to prevent moisture losses. The steam cycle performance was based upon the work discussed in Section I of Part Two. A summary of potassium topping cycle performance estimates is given in Table 89.

Simple Rankine Cycle

The simple potassium Rankine cycle (cycle 1-2-8-2'-3-4'-4-1 in Fig. 85) would not be suitable for topping cycle usage, since, for a specified turbine exhaust moisture content, the potassium condenser temperatures would be too high (see top row, Table 89). Also, for a specified potassium condenser temperature, the moisture content in the turbine would exceed the allowable 12% limit (see second row, Table 89). Therefore, a simple Rankine cycle would be feasible only for a potassium inlet temperature of 1400 F or lower. The cycle efficiency for such a potassium inlet temperature would be 13.8%.

Reheat Rankine Cycle

The potassium reheat cycle (cycle 1-2-8-2'-3-4'-5'-6-7-5-4-1 in Fig. 85) was analyzed for three specified cycle conditions: (1) reheat pressure, P_4', specified, (2) moisture content at high turbine outlet, X_4', specified, and (3) moisture content at low turbine exit, X_7, specified. It would be possible to specify a reheat pressure such that the moisture content limit of 12% at the high turbine exit would also limit the turbine inlet temperature. Similarly, specifying maximum high turbine exit moisture could again limit the maximum turbine inlet temperature, as could specifying the low turbine exit moisture. Thermal efficiency estimates for these potassium cycles under a variety of imposed constraints are given in Table 89.

Regenerative Rankine Cycle

Since the primary interest in the potassium regenerative cycle (cycle 1-2-2'-3-4'-8/5'-5-4-1 in Fig. 85) would be its potentially high performance in the high-temperature range, cycle analyses were made only for specified moisture conditions, i.e., $X_4' = 0.12$ and $X_5 = 0.12$. In either case a minimum temperature of 1800 F for T_3 would be required for the system to be compatible with these constraints. Thermal efficiency estimates for this cycle are also given in Table 89.

System Selection

Based upon the foregoing analysis, two systems were selected for more detailed analyses. One was a low-temperature system which would utilize materials of construction that are currently available; the second was a high-temperature system utilizing technology judged to be attainable within the next two decades. This system would demonstrate very high efficiencies.

Low-Temperature Cycle

An examination of Table 89 shows that in the lower temperature range of T_3 (1400 F to 1600 F), only the reheat cycle with specified reheat pressure would be feasible. Also, the lowest value of T_c, 1040 F, would yield highest efficiency. In addition, 1500 F was considered to be the highest temperature which could be attained using stainless steel. Consequently, these conditions were selected for the low-temperature reheat cycle.

A schematic diagram showing the arrangement of major component for the low-temperature system and a t-s diagram indicating the operating conditions are given in Fig. 87. The performance of the low-temperature system for various potassium inlet temperatures and condenser temperatures is shown in Fig. 88. For the selected potassium inlet condition of 24.7 psia/1500 F, reheat would take place at 6.14 psia/1260 F. The specific power outputs would be 75 Btu/lb and 91.5 Btu/lb, respectively, for the high and low turbines, and the exit moisture content for these turbines would be 10% and 3.2%, respectively. A relatively inexpensive steam system of 2500 psia/1000 F/1000 F with a cycle thermal efficiency of 42.3% was selected for the binary system, primarily for economic reasons. With potassium cycle efficiency of 16.3%, the binary cycle efficiency would be 51.7% and an overall binary plant efficiency of 44.6% would be obtainable. (A boiler efficiency of 88% and a generator efficiency of 98% were assumed for all potassium systems.)

High-Temperature Cycle

Since the high-temperature system would have the benefit of materials technology attained during the next two decades, it was assumed that an advanced material such as TD nickle or columbium, with or without coatings, would be available. Since the potassium pressure level would be relatively low, the heater tubes would not be highly stressed, and metal temperatures on the order of 2100 F should be possible. The corresponding maximum potassium temperature would be approximately 2000 F.

As in the low-temperature case, the lowest possible potassium condenser temperature, 1040 F, was selected. Having specified the potassium inlet temperature and the condenser temperature, the choice of cycle configuration becomes apparent; the regenerative cycle with moisture separation at a specified moisture content of 12% ($X_4' = 0.12$) would have the highest thermal efficiency and thus was selected for further consideration.

A schematic diagram showing the arrangement for the major components and a t-s diagram showing the cycle configuration and operating conditions are given in Fig. 89. For a selected potassium inlet condition of 152.4 psia/2000 F and regeneration by separation of moisture at 22.5 psia/1480 F, the potassium cycle efficiency was estimated at 27.3%. Using a 3500 psia/1000 F/1000 F steam cycle with a thermal efficiency of 44.6%, a binary cycle efficiency of 59.6% or overall binary power plant efficiency of 51.5% would be obtainable.

System Sizing

Although steam systems with potassium topping cycles would have significantly better performance than the reference coal-fired steam station, other advanced-cycle power systems utilizing gas turbines were found to have higher efficiency and offer the potential of lower development and investment cost. Consequently, analysis of potassium systems was discontinued, and the studies of the high-temperature system which were in progress were not completed. Thus, results for the low-temperature system only are presented in the following discussion of equipment size and cost estimated.

The sizes of steam turbine-generation equipment are very much standardized by the power equipment manufacturers and are considered herein. The basic components of the low-cost potassium topping system would include the potassium boiler and reheater unit, the potassium turbines, and the potassium condenser-steam generator. A conceptual design of the low-cost system with a 1000-Mw electric power output would consist of four potassium turbines, each delivering

80 Mw, and a steam system which would deliver 680 Mw. The power split between the potassium and steam systems is based upon a heat balance for a potassium condenser operating at the conditions of the low-temperature cycle, i.e., approximately 0.79 lb/sec of steam at 2500 psia/1000 F/1000 F could be generated by each lb/sec of potassium flow. The flow rates of potassium and steam in the system, therefore, would be 6.2×10^6 lb/hr and 4.0×10^6 lb/hr, respectively.

Potassium Boiler

Preliminary considerations of the transfer characteristics in potassium boilers indicated that the heat transfer would be limited primarily by the relatively low value of heat transfer coefficient (10 to 15 Btu/hr ft^2 F) in the hot gas side. The total heat transfer surface area in the potassium boiler was calculated to be 5.6×10^5 ft^2, including 6.3×10^4 ft^2 for the reheater. Because of the very low potassium pressure level (1.5 to 24.7 psia), thin-wall stainless steel tubes could be used in the potassium boiler. Discussions of the problems associated with liquid metal heat exchangers are contained in Refs. 138 and 139.

Potassium Turbines

The potassium vapor turbine would look very much like a low-pressure steam turbine in a conventional steam station. Four identical turbine units would be used, with each passing approximately 1.55×10^6 lb/hr. In order to keep the physical size of each unit within steam limitations, each turbine would be designed for 1800 rpm and dual-flow exhaust. The mean diameter of the 1800-rpm turbine would be approximately 7 ft. Potassium vapor would be extracted from between the high- and low-pressure sections for reheat. The maximum liquid content in the turbine would be limited to 12% to avoid excessive erosion of turbine blades. The inlet/outlet pressure ratios for the high and low turbines would be 3.5 and 4.7, respectively. Three stages would be required in the high-pressure section and four stages in the low-pressure section.

Potassium Condenser

In the potassium condenser-steam generator, the rate of heat transfer from the condensing potassium to the evaporating water and reheating steam would be approximately 5.3×10^9 Btu/hr. U-shaped tubes, positioned upside down and arranged in cylindrical bundles, which would then be mounted in a casing, would serve as the potassium condenser. The feedwater or reheat steam entering in one leg of the U tubes would be heated while flowing upward and then would return downward to the tube bundle outlet. Because of the high steam pressure in the U tubes, very thick-walled header drums and tubes would be required. Although the potassium temperature would be over 1000 F, its pressure in the condenser would be only about 15 psia; thus, the stress problem of the condenser shell would not be very severe. Because of the very high heat transfer coefficient for the condensing potassium and the boiling water, the overall heat transfer coefficient would also be very high, and the size of potassium condenser-steam generator would be significantly smaller than a conventional steam generator with the same steam rate.

System Economics

Because of the novel component design of a potassium topping cycle, it would be difficult to make an accurate estimate for the system capital cost. Therefore, only approximate cost estimates are presented, and these are for the low-temperature system. The costs of the steam system would be identical to those discussed in Section I of Part Two except for the steam generator. However, as a first approximation, it was assumed that the reduction in steam generator size would be offset by the increased material costs of the tubes and the increased fabrication costs due to the welded construction, which would require X-ray inspection. Thus, the installed cost for a 680-Mw steam power system was estimated to be $100 million. Assuming that the cost of the nickel-base or refractory material to be used in the potassium boiler would be 20 times that of the conventional tubes, but the weight of potassium boiler tubes were to be only about 15% of the weight of conventional boiler tubes, the capital cost of the potassium boiler (with a heat transfer area of 5.6×10^5 ft^2) was estimated to be $33.6 million. Similarily, the estimated cost for the reheater is $3.8 million. The capital cost of potassium turbines was estimated to be $30/kw or $9.6 million for the four of them. The 320-Mw electric generator for the potassium turbines would cost approximately $4.5 million, and the potassium feed pump cost was estimated to cost $400,000. The piping cost was estimated to be $3.5/kw, for a total piping cost of $1.1 million for the potassium system. The total equipment cost for the potassium topping system alone would be the summation of these component costs and would equal approximately $53.0 million. After adding appropriate amounts for engineering, escalation, and interest during construction, the total cost for the potassium system would be approximately $74.1 million. Therefore, the installed cost of a 1000-Mw output binary power generation system would be approximately $174.1 million.

A comparison of the busbar power costs for the potassium system with the reference coal system is given in Fig. 90. It can be seen that only at fuel costs of 22¢/million Btu or lower, well below the probable range of clean fuel costs, would the potassium-steam binary system be competitive with the reference steam station.

BOTTOMING CYCLES

Low-temperature, Rankine-type bottoming cycles could find application in central power stations when used in combination with a steam cycle, a combined gas and steam (COGAS) cycle, or a regenerative gas turbine as illustrated in Fig. 91. The purpose of adding a bottoming cycle to the aforementioned power systems would be to increase overall thermal efficiency, reduce specific cost, or both. In a conventional steam system, the stack gas temperature is usually quite low, and most of the heat rejected by the steam takes place in the condenser at the minimum (sink) temperature; therefore, there would be no chance of improving overall thermal efficiency with a bottoming cycle. However, by eliminating some of the large and expensive low-pressure steam turbine stages and transferring the heat remaining in the exhaust steam to a bottoming cycle employing a high-density working fluid, some savings in turbine size could be achieved, and hopefully the reduction in size would result in lower overall system cost.

A steam-ammonia binary cycle system incorporating these features was analyzed in detail in Ref. 132 and it was shown that substantial cost savings for the turbines might be achieved. In COGAS systems, the gas temperature leaving the steam boiler could, in some configurations, be as high as 600 F. By recovering some of this otherwise waste heat in a bottoming cycle power system, the overall thermal efficiency could be increased. Likewise, in regenerative gas turbine systems, the gas temperature exiting the regenerator could be sufficiently high to warrant investigation of a bottoming system to recover waste heat and improve overall thermal efficiency.

A number of tertiary power cycles, which represent an attempt to combine the low capital cost of the steam-ammonia binary cycle and the high thermal efficiency of the bottomed COGAS cycle, have also been considered. These tertiary cycles are in general highly complex with heat being recovered from both steam and gas turbine combustion gases by an ammonia bottoming cycle.

Steam-Ammonia Bottoming Cycle

A simplified temperature-entropy diagram for a representative steam-ammonia binary cycle is illustrated in Fig. 92. The steam cycle incorporated in this figure corresponds to a second-generation system, i.e., it is supercritical and operates at 3500 psia and 1000 F with reheat to 1000 F. Steam would be expanded in the turbine to a pressure of approximately 30 to 35 psia and then would be condensed at a temperature somewhat above 250 F in an ammonia boiler. Saturated ammonia vapor which would be generated at 1029 psia (70 atm) and 226 F would be expanded to 212 psia and subsequently condensed at 100 F. Although not shown in the figure, the steam cycle would employ four stages of feed heating, and the ammonia cycle would employ two stages.

The above ammonia cycle operating conditions were found to yield near-maximum bottoming cycle thermal efficiency (approximately 11%) when the minimum temperature difference between the steam and ammonia was maintained at a value slightly in excess of 25 F. Although this temperature difference appears to be rather small, it should be noted that the heat transfer coefficients are very high for condensing steam and boiling ammonia and that the size and cost of the heat exchanger required would be relatively small. Furthermore, any attempt to increase this temperature difference would increase the associated thermodynamic irreversibility which is a prime cause of the relatively low thermal efficiencies achieved with this type of binary cycle. For a steam turbine exhaust pressure of 31 psia and the previously mentioned ammonia cycle conditions, the overall thermal efficiency for the binary cycle was estimated to be approximately 41%, which compares to 45% for the conventional steam cycle. These efficiencies do not account for losses in the electric generator, steam boiler, or power system auxiliary requirements, but they do account for the power required to drive the ammonia feed pumps. For a nominal 1000-Mw system, approximately 824 Mw would be generated by the steam system and 176 Mw by the ammonia bottoming system.

In addition to the irreversibility losses associated with heat transfer from the steam to ammonia, the lower steam-ammonia binary cycle efficiency would also be due to the relatively low ammonia turbine efficiency which was estimated to be approximately 75% (isentropic value). Approximately 10 points in ammonia turbine efficiency would be lost due to the relatively high moisture content in the turbine exhaust (approximately 20% for the ammonia cycle conditions previously stated). Although this moisture loss is substantial, any attempt to avoid it by employing ammonia superheat would require lowering the turbine inlet pressure (to maintain a reasonable minimum temperature difference between the steam and ammonia), resulting in an even lower binary cycle efficiency.

Despite the lower efficiency of the steam-ammonia binary cycle relative to that of a comparable conventional steam cycle, cost analyses were conducted to determine whether the capital cost savings attributable to the smaller size of the ammonia turbine relative to the size of the low-pressure steam turbine would more than compensate for the higher operating cost associated with the lower cycle efficiency. This cost analysis was based on the steam-ammonia binary cycle system configuration contained in Ref. 132, but since this Ref. 132 cycle differs only slightly from the cycle configuration described in the previous paragraph, the conclusions from the analysis are considered applicable to both. As a result of this cost analysis, it was concluded that when realistic pricing is used for the turbomachinery and heat transfer components, a conventional steam plant would be slightly less expensive per kilowatt than a steam-ammonia binary system. For example, the apparent turbine cost savings due to a reduction in turbine size, both in the number of stages and diameter, would be offset by higher costs associated with making the ammonia system completely leakproof. Leakproof construction would be necessary because ammonia is toxic, flammable, and relatively expensive. Thus without a lower capital cost, the steam-ammonia binary system could not hope to compete with a conventional steam system.

Steam-Fluorocarbon Bottoming Cycle

A binary-cycle system employing R11 fluorocarbon in the bottoming cycle instead of ammonia was also investigated. The R11 working fluid was selected for analysis because previous experience indicated that a relatively high bottoming cycle efficiency could be achieved at the low temperatures involved. In addition to its favorable thermodynamic properties, saturated R11 vapor would superheat during expansion in a turbine so that there would be no turbine losses due to moisture. Using an R11 turbine efficiency of 85%, the binary cycle efficiency was estimated to be approximately 42% which is one point higher than for the steam-ammonia cycle, but still three points lower than for the conventional steam system. Although no attempt was made to determine the effect on system cost of using R11, it appears that the R11 turbine should be somewhat more expensive than the ammonia turbine. Both fluids have approximately have approximately the same specific volume (approximately 1.4 ft^3/lb) at the turbine exit condition, and both bottoming systems would put out roughly the same power. However, due to the lower specific work of R11 compared to ammonia, the R11 turbine would require a higher flow rate and more exhaust ends. Each exhaust end would require fewer stages (probably only one) for expansion. The R11 bottoming system must also be made leakproof, not because R11 is toxic, but because it is relatively expensive. Almost any organic fluid which might have been selected instead of R11 would also be considered expensive when compared with steam. Since the R11 heat transfer equipment would be similar to that for ammonia, the R11 system would offer no significant capital cost advantage relative to the conventional steam system.

Bottoming Cycles for COGAS and Regenerative Gas Turbine Systems

As previously indicated, the primary purpose for adding a bottoming cycle system to a COGAS or regenerative gas turbine system would be to recover low-temperature heat from the stack gases which would otherwise be wasted and convert it into useful power to improve the thermal cycle efficiency of the overall system. This improvement in overall efficiency would be maximized when the output power of the bottoming cycle is maximized (assuming no change in main power system performance). Selection of the bottoming cycle working fluid is important in maximizing power output because the fluid properties (thermodynamic and physical) can affect both the bottoming cycle efficiency and the heat input to the bottoming system, the product of which constitutes the power output. The bottoming cycle working fluids considered for the COGAS and regenerative gas turbine applications are presented in Table 90 together with their properties. The three fluorocarbons, R11, R12, and R114, are characterized by their low critical temperatures and low latent heats of vaporization. Steam is at the other end of the spectrum having a high critical temperature and a high latent heat of vaporization. Thiophene properties, in general, are between those for the fluorocarbons and steam. Ammonia is a somewhat peculiar working fluid in that it has a relatively low critical temperature but a high latent heat of vaporization. The fluorocarbons are suited only for the low-temperature applications because they are chemically stable up to a temperature of only approximately 325 to 400 F, depending upon the type. R114 is the most stable of those listed in Table 90. Thiophene is chemically stable up to 600 to 700 F and has been shown in previous UARL studies (Ref. 140) to offer some performance advantages over the fluorocarbons or steam. Thiophene is also being seriously considered as the working fluid for Rankine-cycle engines which could power the nonpolluting automobiles of the future. A US patent (No. 3,292,366) covering the use of thiophene as a working fluid in a power generating system was issued on December 20, 1966, and is assigned to United Aircraft Corporation.

Critical temperature is an important working fluid property. Fluids with high critical temperature generally result in high cycle efficiency because heat can be added to the cycle at a high temperature while limiting the inefficiencies associated with adding heat in the superheat region. A comparison of the net cycle efficiencies that could be achieved with the organic working fluids listed in Table 90 are shown in Fig. 93 as a function of turbine inlet temperature. The curves in this figure are based on data taken from Ref. 140. Although the condenser temperatures and turbine efficiencies specified in Fig. 93 differ slightly from the values considered in the current study, the comparison does provide a simple means of making a preliminary selection of the most promising fluid in the group.

Cycle efficiency is only one of the criteria needed to select the best working fluid. The latent heat of vaporization of the working fluid can limit the heat input to the cycle, as exemplified in Fig. 94 for a waste heat source temperature of 480 F, the exit stack temperature could be reduced to 300 F with steam and 200 F with R11, while maintaining a 50 F minimum (pinch) temperature difference between the heat source and working fluid. The temperature drops in the hot gas stream, 180 F for the steam system and 280 F for the R11 system, indicate that, for the conditions specified, approximately 50% more heat could be recovered by the R11 system. It should be pointed out that the bottoming cycles used in the steam binary cycles previously discussed were not faced with this heat input problem to the same degree because the heat source (steam) remained at constant temperature as heat was transferred to the bottoming cycle working fluid.

A comparison of the maximum power that could be generated in a bottoming cycle employing various working fluids is shown in Fig. 95 as a function of maximum (initial) heat source temperature. As previously mentioned, maximum power output is the final parameter which shows the performance advantage of using one working fluid over another in a bottoming cycle utilizing waste heat. The data in Fig. 95 were also taken from Ref. 9. Although the power levels and the conditions specified do not apply directly to the current study, the comparisons in Fig. 95 do provide the basis for selecting the best performing bottoming cycle working fluid for the COGAS and regenerative gas turbine applications currently under consideration. At maximum waste heat source temperatures from approximately 300 to 600 F, the fluorocarbons would yield maximum power. In this region the greater amounts of heat transfer achieved with the low-heat-of-vaporization fluorocarbons outweigh any disadvantages in cycle efficiency. Above 600 F thiophene would yield the highest power output.

Had the comparison in Fig. 95 been extended to waste heat source temperatures beyond 800 F, it would have shown that the power output would be equal for thiophene and steam at approximately 1000 to 1100 F. It may be noted that ammonia does not show a distinct performance advantage in any temperature range. Having selected the best organic working fluid on the basis of the comparison in Fig. 95, maximum power outputs were calculated using the operating conditions specified in the current study and are presented in Fig. 96 in terms of kw/unit heat source flow rate for maximum heat source temperatures ranging from 200 to 1100 F. As noted in Fig. 96, the maximum heat source temperatures for the COGAS and regenerative gas turbine applications range from approximately 200 to 600 F and 750 to 1000 F, respectively. The temperature range from 200 to 350 F applies to COGAS systems employing simple-cycle gas turbines, whereas the range from 350 to 600 F applies to COGAS systems employing gas turbines with compressor intercooling. The data for steam bottoming systems which are also presented in Fig. 96 for comparative purposes are based upon work which is described in detail in Section III.

Although some performance advantage for the organic fluids is indicated for maximum source temperatures between approximately 300 and 1000 F, the magnitude of the improvement over steam is generally rather small. Furthermore, for a 1000-Mw COGAS or regenerative gas turbine system, with a heat rejection temperature above 300 F, the bottoming fluid flow rate would be on the order of 3000 lb/sec, resulting in bottoming cycle output powers in excess of 7.5 Mw. Other studies (Refs. 140, 141, and 142) have indicated that organic fluid bottoming systems do not show significant cost advantages, if any, over steam bottoming systems at this power level and higher.

Thus, there appears to be little benefit in using organic fluids in power systems bottoming COGAS or regenerative gas turbine systems, especially when one considers the undesirable characteristics of these fluids. The fluorocarbons and thiophene are rather expensive (see Table 90) and tend to decompose at low-to-moderate temperatures. In addition, thiophene is flammable. Since steam bottoming systems are included in the investigation of COGAS systems, it was not necessary to analyze them further in this section. As described in the discussion of COGAS systems, there appear to be suitable means of maximizing the performance of COGAS systems without resorting to organic-fluid bottoming cycles. Furthermore, simple waste-heat recovery COGAS systems offer higher performance than do regenerative gas turbines equipped with any combination of power recovery systems.

Tertiary Power Cycles

As suggested in Ref. 143, a number of highly sophisticated tertiary power cycles can be formulated by combining ammonia bottoming cycles with supercharged COGAS cycles. Other bottoming cycle working fluids could be used instead of ammonia, but there would be no significant change to the general characteristics of the overall tertiary cycle system by doing so. However, fluorocarbons and most organic could not be used when high temperatures are encountered because of their low chemical stability.

Examples of tertiary cycles are presented in Figs. 97 and 98. In the cycle illustrated in Fig. 97, high-temperature heat would be added to ammonia to achieve the same level of turbine inlet temperatures that would be used in steam turbines (approximately 1000 F). It may also be noted in Fig. 97 that the low-temperature heat added to the ammonia would come from both steam (exhausting from a noncondensing turbine) and the gas turbine exhaust gases. A variation on this cycle is one in which the high-temperature heat addition to the ammonia would be omitted, and the ammonia turbine inlet temperature would be limited by the saturation temperature corresponding to the steam turbine exhaust as in the previously discussed binary cycle. Another tertiary power cycle, which is illustrated in Fig. 98 in a highly simplified way, would involve direct heat addition to the steam system by means of burning a clean fuel with air. The combustion product gases and the highly superheated steam would be expanded through the same turbine and subsequently cooled against the ammonia in the bottoming cycle. Although the aforementioned tertiary cycles were not analyzed in detail, their merits, if any, can be evaluated qualitatively by applying the results from the studies of related systems.

In the analysis of COGAS systems, it is shown that thermal efficiency would decrease significantly with increased excess firing in the boiler. It should be noted that this trend is due in part to the relatively high gas turbine inlet temperatures assumed in the current study. Although the specific power output would also increase, it appears that the lowest-cost electric power would be generated in high-efficiency low-specific-output power configurations. It is further shown that at low specific power output, the supercharged COGAS system would have approximately the same efficiency as the waste-heat type COGAS system.

There is also some reason to believe that supercharged boilers may not cost less than the waste-heat type, contrary to previous claims. One boiler manufacturer maintains that the high costs of the pressure vessel and special fabrication procedures required by the supercharged boiler would offset the apparent advantage of reduced physical size. Thus, there appears to be no significant advantage to utilizing a supercharged COGAS cycle rather than a waste-heat type in the above-mentioned tertiary cycles.

The cycle shown in Fig. 97 represents an attempt made to overcome some of the inefficiencies characteristic of the previously described steam-ammonia binary cycle. The thermodynamic irreversibilities associated with transferring heat from the condensing steam to ammonia could not be eliminated. However, the power output of the ammonia cycle could be increased by utilizing waste heat in the flue gas for preheating liquid ammonia, leaving the heat from the condensing steam to vaporize more pounds of ammonia. Increased power output from the ammonia cycle in turn would lead to an improvement in overall cycle efficiency. By utilizing some high-temperature heat from the gas turbine supercharged boiler combustion chamber, the ammonia could be superheated before expansion in the turbine, thus avoiding the moisture losses which would be present in the saturated ammonia bottoming cycle used in the steam-ammonia binary cycle.

Although the ammonia turbine required in this tertiary cycle would differ from that required in the steam-binary cycle, their comparison with corresponding steam turbines should be similar, i.e., the cost of making the ammonia system leak-proof would outweigh the apparent cost savings in the turbine due to its reduced physical size. Thus, tertiary cycles similar to that illustrated in Fig. 97 apparently offer no significant advantage over a waste-heat COGAS cycle designed for equivalent stack gas temperatures.

The tertiary cycle utilizing some direct heat addition to the steam (Fig. 98) could be thermodynamically equivalent to that illustrated in Fig. 97 minus the high-temperature heat addition feature to the ammonia bottoming cycle. However, with direct heat addition the ammonia bottoming cycle would be required to convert the heat remaining in the steam exhausting from the turbine into useful work if there is to be any hope of translating the higher steam temperatures into higher cycle efficiencies. In this arrangement, the steam could be fully expanded to 2 in. Hg by practical means because of the presence of noncondensible gases formed by the combustion of fuel and air. The ammonia bottoming cycle would not be needed from a performance standpoint if pure hydrogen fuel and oxygen were available at low cost, which they are not. Hydrogen and oxygen could be burned in stoichiometric proportions forming only steam which could be expanded to 2 in. Hg by condensing at a temperature of 100 F. The direct heat feature of this cycle would contribute little to increasing overall cycle efficiency because heat added to this way would be recovered only at the efficiency of the combined steam-ammonia cycle and would not contribute to the gas turbine performance.

Furthermore, the efficiency of the steam-ammonia cycle would not increase significantly because the beneficial effect of increased steam temperature on one end of the cycle would be offset on the other end by reduced output of the ammonia bottoming cycle. As pointed out in Ref. 143, any increase in direct heat addition would lower the dewpoint temperature of the steam-combustion products mixture which would limit the ammonia bottoming cycle temperature and efficiency. In addition to generating power, ammonia in the bottoming cycle would provide the means for condensing out steam for boiler feed purposes. It should be noted that in this direct heat addition tertiary cycle the steam temperature entering the turbine would be limited by considerations other than those dealing with structural design. As may be noted in Fig. 98, one of the disadvantages of the direct heat addition feature is that both the fuel and air used for this purpose must be supplied at the extremely high pressure corresponding to the steam throttle requirements which might be on the order of 2400 psia or higher.

The tertiary power cycles discussed in the previous paragraphs represent an attempt to achieve high cycle thermal efficiency whereby highly sophisticated and supposedly highly efficient steam and ammonia bottoming systems would be employed to recover waste heat from gas turbines. This approach might have merit from a performance standpoint at gas turbine inlet temperatures limited to approximately 1600 to 1700 F. However, at the high temperatures (2200 to 3100 F) considered in this study, high cycle thermal efficiency could be achieved with a much simpler waste-heat COGAS cycle. Because of its simplicity, the COGAS system should also cost less than the tertiary systems.

Section V - Closed-Cycle Gas Turbine Power Systems

This section contains the results of an investigation to evaluate the technical and economic feasibility of closed-cycle gas turbine power systems. A wide variety of working fluids was considered for these power systems, although helium, carbon dioxide, and sulfur dioxide received principal attention. Descriptions are presented of various cycle configurations incorporating intercooling, regeneration, and reheat. Parametric performance calculations were made for each of these cycles in order to determine the specific cycle configurations which looked most promising for each working fluid. Critical design problems which would limit the design of high-temperature, closed-cycle gas turbine systems are also described. Two operating temperature levels were selected in order to cover a wide range of system performance and economic characteristics. The lower temperature level was considered to represent technology currently available, although possibly not yet reduced to commercial practice, whereas the higher temperature level was considered to represent technology judged to be attainable for commercial use within the next 20 years. Capital cost estimates were prepared for those closed-cycle systems that were judged to have the best potential for producing low-cost power. Electric power costs were estimated for these selected systems as a function of the net cost of desulfurized fuel delivered to the power system. The results of these analyses indicate that closed-cycle systems are not competitive with conventional steam power systems or with advanced-cycle power systems based on open-cycle gas turbines.

CHARACTERISTICS OF WORKING FLUIDS

The concept of the closed-cycle gas turbine was developed in Switzerland and first patented in 1935 (Ref. 107). Air was originally considered as the working fluid. A pilot plant demonstrating this concept was begun in 1936, and in 1939 a 2000-kw system went on line, While this concept has not received worldwide acceptance, there are approximately one dozen closed-cycle plants operating in Europe and Asia at power levels up to 17 Mw. These systems are fired with a variety of fuels, such as bituminous coal, mine gas, and natural gas. The current application of closed-cycle gas turbines has been limited to industrial applications. The advent of the gas-cooled nuclear reactor, however, has rekindled interest in the use of closed-cycle systems in large commercial central power stations. A 25-Mw prototype gas-cooled reactor closed-cycle gas turbine power system in which helium comprises both the reactor coolant and the working fluid for the power cycle is presently being constructed (Ref. 145). Closed-cycle gas turbine power systems utilizing carbon dioxide, sulfur dioxide, and other potential working fluids have also been considered recently (Refs. 139, and 146 through 154).

Closed-cycle gas turbine power systems are external-combustion systems, i.e., heat released by combustion of fossil fuels would be transferred to the working fluid in a heat exchanger. Because of this method of heating the working fluid, the maximum metal temperature in the heat exchanger would be several hundred degrees higher than the turbine inlet temperature. These systems are quite different from the more familiar open-cycle gas turbine power systems in which the products of combustion comprise the power cycle working fluid. Since closed-cycle systems are, as the name implies, closed, the choice of working fluid is not limited to the more common and less-expensive fluids but could be any compressible fluid having desirable properties for closed-cycle applications. Thus, the selection of working fluid is an important aspect of closed-cycle gas turbine power system analyses.

A wide variety of working fluids is available for use in closed-cycle gas turbine power systems. The characteristics of importance in selecting a working fluid for these power systems are chemical inertness, heat transfer and fluid flow characteristics, cycle performance, turbomachinery design parameters, and availability. Previous investigations at UARL (Ref. 139) and elsewhere (Refs. 145 and 5) have indicated that the inert gases would provide most of these desirable characteristics. In addition, carbon dioxide and sulfur dioxide were also examined because it appeared that several high-performance cycle configurations could be devised for use with these working fluids.

Inert Gases

Since the critical temperatures of inert gases are extremely low, these gases behave essentially as perfect gases and would be used in conventional Brayton cycles. The inert gases specifically analyzed were helium, argon, neon, and mixtures of helium and argon. Properties of these gases are given in Table 91 with air and nitrogen included for comparison. The use of neon was not considered seriously because it is more expensive than both helium and argon and also because a mixture of 45% argon and 55% helium would have essentially the same thermodynamic characteristics as neon.

In order to assess the relative merits of these inert gases in closed-cycle systems, simplified performance analyses were made assuming that the fluids behaved as perfect gases, i.e., the specific heats were assumed to be independent of temperature. The results of these simplified analyses are shown in Fig. 99 where it can be seen that with equal temperature ratio, regenerator effectiveness, component efficiencies, and pressure loss, the maximum cycle efficiency is only a weak function of specific heat ratio for each fluid. Similarly, the compressor pressure ratio required to attain maximum cycle efficiency is also a relatively weak function of specific heat ratio. Since the specific heat ratios for inert gases do not differ greatly, as shown in Table 91, the maximum cycle efficiencies for these fluids would be essentially the same. Consequently, the selection of inert gas working fluids must be based upon considerations of the cost characteristics of heat transfer equipment and turbomachinery.

Using simplified calculation techniques, preliminary estimates were made for heat exchanger and turbomachinery design characteristics for closed-cycle gas turbine power systems utilizing the gases shown in Table 91. The design criterion adopted for these systems was maximum cycle thermal efficiency. In addition, equal circumferential velocity was assumed for all turbomachinery designs studied. Since the maximum cycle efficiency for these inert gases would occur at approximately the same isentropic temperature ratio for the compression process, the number of compressor stages would be a direct function of the specific heat of each working fluid. A helium compressor, therefore, would require approximately 10 times as many stages as an argon compressor and five times as many stages as an air or nitrogen compressor for the same circumferential blade velocity. Counterbalancing this disadvantage of helium turbomachinery are the excellent heat transfer characteristics of this gas as indicated in Fig. 100.

The curves shown in this figure indicate the effect of pressure level on the film heat transfer coefficient of several gases at heat exchanger core pressure drops of 1% and 3% of the inlet pressure. Helium is clearly the most effective heat transfer medium. In fact, for the simple regenerative-cycle configuration, a helium heat exchanger would require only about one-fourth the surface area of an argon heat exchanger and one-half that of an air or nitrogen heat exchanger. The results of these preliminary heat exchanger and turbomachinery design analyses are summarized in the lower half of Table 91.

The Table 91 results clearly show that helium would require less heat transfer surface area but that argon would require fewer stages of turbomachinery. In order to evaluate the relative costs of heat exchanger and turbomachinery components, preliminary cost estimates for these items were made. These preliminary cost studies showed that, even though the turbomachinery costs are a direct function of the number of stages, the heat exchangers are a much more significant portion of the total system cost and thus are the dominant factor in working fluid selection. Since helium would result in the lowest heat exchanger cost, it was selected as the inert gas working fluid for the closed-cycle gas turbine power system. The physical properties of helium are summarized in Ref. 155.

Carbon Dioxide

The critical point of carbon dioxide is 87.5 F and 1070 psia. Unlike the inert gases considered in the preceding paragraphs, which would operate well above their critical points and behave essentially as perfect gases, the relatively high critical temperature of carbon dioxide would introduce appreciable real gas effects. Since the critical temperature of carbon dioxide is just slightly lower than the minimum cycle temperature of approximately 100 F usually attained in central power stations, condensing cycles would not be possible. However, a cycle using carbon dioxide as the working fluid has the potential of using a supercritical fluid, as opposed to a perfect gas, in that portion of the cycle where the fluid pressure must be raised. This supercritical fluid could be pumped, similar to the pumping of water in supercritical steam cycles, instead of being compressed as in Brayton cycles. A discussion of the carbon dioxide cycle configurations investigated is included in the following discussion of performance analyses. The physical properties of carbon dioxide are summarized in Refs. 156 and 157.

Sulfur Dioxide

In many respects, sulfur dioxide would resemble steam as a power cycle working fluid. Its critical point is 315 F and 1143 psia. Its critical temperature is well above the minimum cycle temperature of 100 F, so that it would be possible to use sulfur dioxide in a classic Rankine cycle configuration. There are advantages, however, of combining the Rankine and Brayton cycles as discussed in the following section on performance analysis. The physical properties of sulfur dioxide, some of which were derived for the purposes of this study, are summarized in Appendix H.

PERFORMANCE ANALYSES

The following paragraphs contain discussions of parametric performance analyses for helium, carbon dioxide, and sulfur dioxide cycles. These analyses led to selection of specific cycle configurations for each working fluid. The following discussions deal exclusively with the thermal efficiency of the closed-cycle gas turbine and do not include stack losses or thermal losses during fuel conversion.

Helium Power Cycles

Because of previous work on closed-cycle gas turbine power systems (Ref. 139), it was known that compressor intercooling, regeneration, and reheat could be used to enhance the performance of closed Brayton cycles. In order to evaluate the numerous cycle alternatives which were available, a digital computer program was developed to calculate cycle thermal efficiency for varying compressor pressure ratios and turbine inlet temperatures. Performance estimates made using this computer program and the Ref. 3 results are summarized in Figs. 101 through 104.

Figure 101 depicts the performance of a simple cycle in addition to cycles incorporating a single intercooler, an intercooler plus regeneration, an intercooler plus regeneration plus reheat, and two intercoolers plus regeneration plus reheat. These cycle calculations are all based on 1600 F turbine inlet temperature and 90 F compressor inlet temperature. The cycle pressure drops depicted in this figure were estimated to correspond to the number of components required for each cycle. The simple cycle would have a maximum efficiency of approximately 31%. This could be increased by a few percentage points by incorporating intercooling, but a more-significant improvement in performance would be possible by using both intercooling and regeneration as shown in the figure. The amount of heat that could be recovered in a regenerator is related to the difference in turbine exhaust gas and compressor exit temperatures. Regeneration would be most effective when this temperature difference is large or when the compressor pressure ratio is low. The performance of a regenerator is represented by its effectiveness, which is the ratio of the heat actually recovered from the turbine exhaust gases to that heat which is theoretically recoverable. The variation in cycle thermal efficiency with regenerator effectiveness is depicted in Fig. 102 for a nonreheat cycle with one or two stages of intercooling. There are, however, both economic and practical limits to the maximum allowable regenerator effectiveness because the regenerator heat transfer surface area and cost are related to effectiveness by an exponential function. This function is represented by the curve in Fig. 105 for operating conditions which are representative of the power cycles considered. The results in Fig. 105 were normalized with respect to a regenerator effectiveness of 85%, although subsequent analyses indicated a slightly higher regenerator effectiveness would lead to lower power costs.

Since the specific work done by a turbine is a direct function of the mean gas temperature during the expansion, cycle efficiency would be increased by reheating the working fluid at intermediate points during the expansion. A convenient location for insertion of a reheater is between the high-pressure compressor drive turbine and the low-pressure power turbines as depicted in the cycle configuration shown in the bottom of Fig. 101. By use of this cycle configuration, a net cycle thermal efficiency of approximately 48% would be realized, as represented by the uppermost curve in Fig. 100. This efficiency is approximately three percentage points higher than the cycle thermal efficiency of the best conventional steam station now operating.

The effect of turbine inlet temperature on cycle thermal efficiency is depicted by the curves in Fig. 103. Cycles with turbine inlet temperatures of 1000 F to 1200 F should be within the current state of the art. However, these cycles demonstrate undesirably low cycle efficiency. Cycles with turbine inlet temperatures above 1800 F appear to be capable of thermal efficiencies in excess of 50%. These cycles, however, would present some rather serious materials problems as discussed in subsequent paragraphs.

The cycle pressure drops depicted in Fig. 101 are considered to represent relatively conservative estimates for this parameter. Previous estimates of helium power systems (Ref. 139) showed that a 5% decrease in cycle pressure drop would result in approximately a 2 point increase in cycle efficiency. In order to achieve reduced cycle pressure drop, however, the flow velocities in the heat exchanger must be decreased, resulting in lower heat transfer coefficients and larger heat exchangers. These larger heat exchangers, obviously, would be more expensive than heat exchangers designed for low pressure drop, so that selection of final cycle pressure drop would require detailed cost optimization studies. Since closed-cycle gas turbine power systems were found to be less attractive than power systems incorporating open-cycle gas turbines, these optimization studies were not conducted.

In a closed-cycle gas turbine system the working fluid would always remain in the gaseous phase, where the temperature and pressure are independent properties. Therefore, the maximum pressure level would not be dictated by selection of the operating temperature but could be increased to levels limited only by the strength of component materials or by component design restrictions. It would generally be desirable to operate at a high pressure level because this would result in smaller physical dimensions for the turbomachinery leading to reduced blade stresses. Exploratory performance calculations to determine the effect of operating pressure level on cycle performance, reported in Ref. 102 are presented in Fig. 104. The cycle thermal efficiency would decrease slightly as system design pressure is increased above approximately 1000 psia because the enthalpy of the gas in the region of the compression process in the cycle would become a function of pressure as well as temperature. The compressor work for a constant pressure ratio then would increase as maximum pressure increases resulting in reduced net work output and reduced thermal efficiency. As a result of these exploratory studies, operating pressure levels for helium systems were selected to be between approximately 700 and 1000 psia for subsequent analytical studies.

Carbon Dioxide Power Cycles

Since most practical carbon dioxide cycles would involve operation relatively near the critical point (87.5 F and 1070 psia), analysis of these cycles must include real gas effects. Thorough discussions of these effects are given in Refs. 148 and 150. Since the critical temperature of carbon dioxide is below the minimum cycle temperature assumed for the present study, all carbon dioxide cycles considered herein are of the noncondensing type. Five different noncondensing cycles were evaluated for the present study (see Fig. 106). These cycles are, from lowest to highest performance, (1) the Feher cycle (E in Fig. 106); (2) the Feher cycle with split flow (D in Fig. 106); (3) the Brayton cycle (C in Fig. 106); (4) a combined Feher-Brayton cycle (A in Fig. 106); and (5) a combined Feher-Brayton cycle with reheat (B in Fig. 106).

The Feher cycle (Refs. 151, 152, and 153), which would operate entirely at supercritical pressures, was created to take advantage of the reduction in gas specific volume at elevated pressure and its commensurate reduction in compressor work. However, unless excessively high turbine inlet pressures could be used, the expansion ratio across the turbine would be limited, resulting in relatively low turbine work and low cycle efficiency. A second drawback, which affects all remaining carbon dioxide cycles considered except the Brayton cycle, involves design of a practical regenerator which takes into account the real gas effects of carbon dioxide. As is the case in helium cycles, regeneration would be necessary in all carbon dioxide cycles in order to achieve high cycle thermal efficiency.

Another cycle configuration investigated involves splitting the hot and cold gas streams as shown by Cycle D in Fig. 106. In this cycle the cold gas flow stream exiting the low-temperature regenerator would be split. One portion of this split stream would be compressed and then fed directly into the high-pressure side of the high-temperature regenerator. The remaining portion of the split stream would be precooled, compressed, heated in the low-temperature regenerator, and then passed into the high-temperature regenerator. Sufficient heat must be transferred in the low-temperature regenerator to raise the temperature of the high-pressure carbon dioxide to the same temperature level as that of the carbon dioxide which was compressed.

The straight Brayton cycle (Cycle C in Fig. 106) would not be limited to the relatively low overall cycle pressure ratio that the Feher cycle would be limited to, nor would the regenerator have to be designed for working fluids whose properties might vary drastically in the vicinity of the critical point. The higher turbine expansion ratio of a Brayton cycle would enable this cycle to achieve significantly higher performance than the Feher cycle.

By combining the features of the Brayton cycle with the features of the split-stream Feher cycle, the high-performance cycle designated A in Fig. 106 would be possible. As shown in this figure, cycle thermal efficiencies in excess of 40% would be possible for turbine inlet temperatures on the order of 1400 F. By incorporating reheat, as shown in Cycle B, 40% thermal efficiencies could be achieved for turbine inlet temperatures as low as 1200 F. The effect of regenerator effectiveness on cycle thermal efficiency for these combined Feher-Brayton cycles is depicted in Fig. 107. For those cycles with reheat, thermal efficiencies between 40 and 45% would be possible for turbine inlet temperatures of 1200 F, depending on regenerator effectiveness. For nonreheat cycles, thermal efficiencies between approximately 42 and 50% would be possible for turbine inlet temperatures on the order of 1600 F.

Sulfur Dioxide Power Cycles

Because the critical temperature of sulfur dioxide is greater than the 100 F minimum cycle temperature assumed in the present study, cycles employing partial or complete condensation of sulfur dioxide would be possible. Cycles which reject heat through condensation would be thermodynamically superior to cycles such as the Brayton and Feher cycles because the condensing cycle would have a lower average temperature at which heat would be rejected from the cycle. In addition, the fluids which would be condensed could then be raised to high pressure by liquid pumping which would require much less power than compression of gas. Furthermore, heat rejection through a condensation process would be advantageous because of the inherently higher heat transfer coefficients which would permit design of compact heat transfer equipment.

The sulfur dioxide cycle configurations investigated range from a simple Rankine cycle to a complex partial-condensation supercritical cycle with reheat, regeneration, and intercooling. The simple subcritical Rankine cycle (Cycle 1 in Fig. 108) would be uneconomical when compared with a similar steam cycle because its thermal efficiency would be limited to approximately 22%. A supercritical Rankine cycle using sulfur dioxide (Cycle 2 in Fig. 108) would have considerably higher thermal efficiency than the subcritical sulfur dioxide Rankine cycle. By adding regeneration and reheat (Cycles 3 and 4, respectively, in Fig. 108) to the supercritical cycle, thermal efficiencies in excess of 40% could be achieved as shown in the figure. Parametric performance estimates for Cycle 4, the supercritical cycle with reheat and regeneration, are depicted in Fig. 109 for several values of maximum cycle pressure and regenerator effectiveness.

A partial-condensation sulfur dioxide cycle with regeneration is shown as Cycle 5 in Fig. 108. In this cycle, only a fraction of the sulfur dioxide would be condensed. The remainder would exit the regenerator and be compressed to heater inlet conditions. The major disadvantage of this cycle would be the severe constraints placed on the turbine inlet properties and regenerator effectiveness required. For an inlet pressure of 3500 psia or higher and an inlet temperature of 2200 F, a regenerator effectiveness of 0.9 or greater would be required in order for this cycle to operate. For lower turbine inlet temperatures, higher regenerator effectivenesses would be required. For a turbine inlet pressure of 2500 psia, the minimum generator effectivenesses required would be 94% and 88%, respectively, for inlet temperatures of 1800 F and 2100 F. Because of the excessively high turbine inlet temperatures, pressures, and regenerator effectivenesses, this type of cycle could never be feasible in commercial installations.

By adding reheat to a partial-condensation sulfur dioxide cycle, the disadvantages described in the preceding paragraph could be alleviated. Such a cycle is shown as Cycle 6 in Fig. 108. For example, the required regenerator effectiveness at 3500 psia/1000 F would be 88% rather than the 94% as would be required for a nonreheat cycle. Cycle thermal efficiency estimates for this cycle, depicted in Fig. 110, approach 60% for 3500 psia cycles.

HIGH-TEMPERATURE HEAT EXCHANGER DESIGN CONSIDERATIONS

Preliminary design studies of closed-cycle gas turbine power systems indicated that there should be no critical design limitations associated with the turbomachinery. The only turbomachinery design problem appeared to be high stress levels in the long turbine blades of the low-pressure turbine section. These problems could be circumvented, however, by employing double-flow, low-pressure turbine sections or multiple turbine units, as indicated in the subsequent summary of design characteristics.

The high-temperature heat exchangers required for closed-cycle systems, however, were found to present rather serious design problems. Two specific design problems were considered. First, because the working fluid employed in closed-cycle systems would be relatively expensive and/or toxic, welded construction must be employed in order to prevent leakage of these working fluids to the environment. In the case of sulfur dioxide systems, leakage of water vapor into the system would have to be avoided in order to prevent formation of sulfuric acid within the system and subsequent acid corrosion of heat exchanger tubes (Ref. 154). A much more serious design problem, however, exists because closed-cycle systems must be operated at high temperature levels in order to achieve high cycle efficiency. Conventional heat exchanger materials are known to exhibit rather poor stress-temperature characteristics at elevated temperature levels, and materials suitable for use at high temperature are known to be relatively expensive. This problem of materials limitation and how it would affect design of the high-temperature heaters and regenerators for closed-cycle systems is discussed in the following paragraphs.

High-Temperature Heater

The high-temperature heater is analogous to the boiler in a steam system. It is essentially a fired heat exchanger in which the desulfurized fuel would be burned to produce the high-grade heat required to raise the working fluid temperature to the desired level. The major difference between this fluid heater for closed-cycle gas turbine systems and conventional boilers arises from the different heat transfer coefficients on the inside of the heater tubes. The heat transfer coefficient on the inside of the fluid heater tubes would be lower than the coefficient on the outside, causing the majority of the temperature drop between the combustion gases and the working fluid to occur on the inside of the tube. As a result, the tube wall temperature would be higher than would be the case if water were being heated as in a conventional boiler.

Tube burnout, therefore, could be a serious problem in fired heaters so that careful estimates must be made of the expected tube temperature profiles in order to design long-lifetime heaters. A method for making this estimate is given in Ref. 158. This method is based upon the solution of an integral equation which gives the ratio of local-to-mean heat flux in the heater. From this heat flux ratio, the local fluid temperature and tube wall temperature could be readily calculated. A digital computer program was developed to solve this integral equation. This computer program utilizes the heat transfer relationships given in Ref. 146 in order to establish the temperature distribution along the length of the heater tube. Heat flux and temperature distributions for a typical fluid heater configuration are given in Fig. 111. It is interesting to note that the point of maximum wall temperature would not coincide with the point of maximum fluid temperature, but would be strongly dependent on the local heat flux. Based on the data presented in Ref. 146, a mean heat flux of 40,000 Btu/hr/ft^2 was used for helium heaters. Corresponding heat fluxes of 24,000 and 10,000 Btu/hr/ft^2 were assumed for carbon dioxide and sulfur dioxide heaters, respectively.

Preliminary calculations made using this computer program showed that the maximum tube wall temperatures in the fired heaters would be approximately 200 F higher than the fluid temperature inside the tube. Since the performance results described in the preceding paragraph indicate that maximum working fluid temperatures in excess of 1600 F would be desirable, maximum metal temperatures in excess of 1800 F would also be desirable.

Because of the need for high-temperature materials, a materials survey was conducted based on currently available literature (Refs. 159 through 163), previous UARL experience (Ref. 164),and discussions with UARL materials experts. The compositions of a variety of current and prospective heat exchanger alloys surveyed are presented in Table 92. The maximum allowable metal stresses for most of these materials, based on the ASME Boiler Code Specifications (Ref. 165), are presented in Fig. 112, as a function of metal temperature. From this figure it can be seen that 1800 F would be the maximum metal temperature that would allow reasonable stress levels (2000 to 5000 psi) to be attained. Based on this 1800 F temperature limit, 1600 F appears to be the maximum allowable working fluid temperature for closed-cycle gas turbine systems. Advanced columbium and columbium F48 would permit much higher operating metal temperatures, but these materials were considered to be much too expensive for use in commercial heat exchangers. TD nickel was assumed for all high-temperature heat exchanger surfaces required to operate between approximately 1400 F and 1800 F. Those heater surfaces required to operate between approximately 1100 F and 1400 F were assumed to be constructed of austenitic stainless steel such as Type SA-312-TP-316.

The use of carbon dioxide in high-temperature gas turbine cycles would present a unique problem because minor dissociation of carbon dioxide would occur at temperatures on the order of 1600 F and below, causing carbonization and carbide precipitation from ferrous-based materials of construction. A study of carbon dioxide as a reactor coolant (Ref. 140) revealed that the structural properties of mild steel become unacceptable at temperatures approaching 1000 F. Low-alloy ferritic steels become marginal at 1000 F and generally unacceptable beyond approximately 1050 F. Austenitic stainless steels, such as Type 304 and 316, are subject to severe chromium carbide precipitation at grain boundaries when exposed to temperatures above 1050 F. Certain high-chromium and nickel-base alloys, however, appear to be acceptable for use at high temperatures when exposed to carbon dioxide environments (see Fig. 113). As shown in this figure, 25/30 stainless steel or 20/20 stainless steel stabilized with magnesium or silicon appear to have reasonably low corrosion rates in carbon dioxide and should be suitable for use in high-temperature heaters.

The literature on the design of fired, high-temperature heaters for use with closed-cycle gas turbine systems is not extensive. For the purposes of the study described herein, high-temperature heater designs were based on information contained in Refs. 166 through 171. The furnace would consist of numerous tube banks suspended, much like superheater platents in conventional boilers, to form a honeycomb array of hexagonal cells. The hexagonal configuration would permit each cell to approach the ideal circular configuration which results in the best radiation heat transfer characteristics (see Refs. 170 and 171). Individual burners would be located at the top center of each cell and would be downward fired. The temperature of the combustion gases would be reduced from approximaely 3500 F to 2000 F within the furnace before being passed into the convection section. Analysis of the full-load temperature distributions for the tube banks indicated that approximately 10% of the tube material would have to be TD nickel in order to withstand the high tube wall temperature. However, at part-load conditions the point of maximum heat flux and, therefore, the point of maximum tube wall temperature would shift upward toward the burner. This effect would be mitigated in part by the fact that part-load heat fluxes would be reduced so that the maximum tube wall temperatures would also be reduced somewhat. Nevertheless, in order to accommodate part-load operation it was estimated that approximately 30% of the furnace tube walls would have to be made of TD nickel. The remaining tube wall surface would be made from stainless steel.

Approximately 40% of the total heat transferred in the heater would be transferred in the convection section. This section would consist of a series of cross-flow, finned-tube heat exchangers with the combustion gases on the outside and the working fluid on the inside. Based on data contained in Refs. 169 and 172, the average heat flux for helium systems would be approximately 7500 Btu/hr/ft^2. Average heat fluxes in carbon dioxide and sulfur dioxide convection sections would be correspondingly lower.

Since the TD nickel that would be required for use in 1600 F cycles is relatively expensive, a system with a lower-temperature cycle which would have lower performance but could be constructed of less costly materials was also examined. A maximum cycle temperature of 1200 F was selected for this low-temperature cycle so that stainless steel could be used in the high-temperature section of the fluid heater.

Regenerator

A review of the literature concerning regenerators (Refs. 139, 146, and 172 through 176) was made to determine the approach to be used in designing the regenerator. As a result of this literature review, it was concluded that the regenerator computer program developed for the Ref. 139 study would be suitable for use in closed-cycle gas turbine systems. This computer program, which was based on heat exchanger data obtained in Ref. 172, provides detailed temperature, pressure drop, surface area, and other design information for a wide variety of regenerator configurations and materials. Example design data from this program for one representative helium regenerator are presented in Fig. 114.

All regenerators for the carbon dioxide and sulfur dioxide systems as well as the low-temperature sections of helium regenerators could be fabricated from a low-cost carbon steel such as SA-106-B. The high-temperature section of the helium regenerator would require a low-alloy steel such as SA-199-T11. As for the high-temperature fired heaters, leakage from regenerators could pose serious problems necessitating the use of welded construction. Both multipass, shell-and-tube-type regenerators and cross-flow, plate-finned type regenerator designs were utilized depending on the fluid pressure level and required regenerator effectiveness.

CHARACTERISTICS OF SELECTED CLOSED-CYCLE GAS TURBINE POWER SYSTEMS

The following paragraphs summarize the design, performance, and economic characteristics of selected helium, carbon dioxide, and sulfur dioxide power systems. These systems were selected in order to cover as wide a range of system performance and economics as technology would allow. Two operating temperature levels were selected based on the high-temperature heat exchanger design considerations discussed previously. The low temperature level was selected to be 1200 F because this appeared to be the maximum temperature which could be achieved using stainless steel. The high temperature level was selected to be 1600 F because this appeared to be the maximum temperature that could be achieved using advanced nickel-based alloys. The high-temperature helium and sulfur dioxide power systems were designed to utilize technology to the fullest in order to attain the highest cycle thermal efficiencies for given operating conditions, whereas the low-temperature helium and sulfur dioxide systems were designed for low cost. Both the high- and low-temperature carbon dioxide systems were designed to have a cycle thermal efficiency equivalent to that for more advanced steam systems (approximately 45%) so that system costs could be compared at approximately equal performance levels.

Design Characteristics

Simplified schematic diagrams for the selected high- and low-temperature helium, carbon dioxide, and sulfur dioxide power systems are depicted in Figs. 115, 116, and 117, respectively. The operating conditions for these power systems are summarized in the upper portions of Tables 93, 94, and 95, respectively. The high-temperature helium system would require two 500-Mw units in order to attain 1000 Mw, whereas the low-temperature helium system would require four 250-Mw units. Both high- and low-temperature carbon dioxide and sulfur dioxide power systems would require two 500-Mw units. Estimated turbomachinery design specifications are also presented in these tables for each power system. These design specifications include shaft rotational speed, number of stages, mean diameter, and isentropic efficiencies for the compressors and turbines. Specifications for the heat exchangers required for the helium, carbon dioxide, and sulfur dioxide power systems are presented in Tables 96, 97, and 98, respectively. These specifications include estimated heat transfer surface areas for the furnace, convection section, and air preheater section of the fired heateres. Estimated heat transfer areas for the regenerators and condensers are also shown in these tables.

Performance Characteristics

Cycle performance characteristics for the helium, carbon dioxide, and sulfur dioxide cycles are represented by the temperature-entropy diagrams in Figs. 118 through 123. Also shown in these figures are estimated temperatures, pressures, and enthalpies for intermediate cycle conditions. Thermal efficiencies for these cycles range from 42.3% for the low-temperature helium cycle to 57.7% for the high-temperature, partially condensing sulfur dioxide cycle.

The mechanical-electrical efficiency for the turbine-generator sets was estimated to be approximately 97%, and the efficiency of the fired heaters was estimated to be approximately 88% (corresponding to 340 F stack temperature). The resulting net power system efficiencies are shown in Tables 93, 94, and 95, to range from 36.1% for the low-temperature helium system to 49.2% for the high-temperature sulfur dioxide system. This latter efficiency is approximately 30% greater than could be achieved with conventional steam power systems.

Economics

Capital cost estimates for the selected helium, carbon dioxide, and sulfur dioxide power systems are presented in Tables 99, 100, and 101, respectively. These capital cost estimates are tabulated according to standard FPC account numbers and include estimates for engineering, design, construction supervision, escalation during construction, and interest during construction. These estimates are based upon such sources as Refs. 167, 174, and 177, the gas turbine costing procedure described in Appendix F, and established procedures for estimating costs of erection, buildings, and structures, and miscellaneous equipment. The costs of operating labor, maintenance, and supplies were estimated to be 0.5 mills/kwhr for all of the closed-cycle gas turbine power systems. Electric power cost estimates for helium, carbon dioxide, and sulfur dioxide power systems are presented in Figs. 124, 125, and 126, respectively, in terms of the cost of fuel supplied to the power system. The estimated probable range of clean fuel costs is also depicted on each of these figures along with the estimated cost of electricity from conventional coal-fired steam power stations. For the entire range of probable clean fuel costs, the estimated busbar power costs for each of these closed-cycle power systems is seen to be significantly greater than the estimated power costs for conventional coal-fired steam stations.

Because closed-cycle gas turbine power systems do not appear to be competitive with conventional steam systems or power systems incorporating open-cycle gas turbines, consideration of closed-cycle gas turbine systems was not continued, and no attempt was made to fully integrate this type of power system with selected fuel conversion processes.

TABLE 41

OPERATING CONDITIONS OF FOSSIL-FUELED STEAM POWER SYSTEMS

Fuel	No. Units and Nominal Size (Mw)	Steam Conditions	Time
Untreated Oil	2-500	2400/1000/1000	1970
	1-1000	3500/1000/1000	1980
	1-1000	3500/1000/1000	1990
Untreated Coal	2-500	2400/1000/1000	1970
	1-1000	3500/1000/1000	1980
	1-1000	3500/1000/1000	1990
Desulfurized Oil	2-500	2400/1000/1000	1970
	1-1000	3500/1000/1000	1980
Gasified Coal or Oil	2-500	2400/1000/1000	1970
	1-1000	3500/1000/1000	1980
	1-1000	3500/1000/1000	1990

TABLE 42

CHARACTERISTICS OF FIRST-GENERATION CONVENTIONAL FOSSIL-FUELED STEAM POWER STATIONS

Oil-Fired

Two 500-Mw units
2400 psig, 1000 F/1000 F single reheat steam cycle
345 kv output from transformers
Natural draft cooling towers required
Stack height is 600 feet
Small diameter cyclones used for dust removal
Location is in northeast United States
Architectural treatment to yield an attractive plant
Oil can be brought in by small tanker
Alternative costs developed for two construction sites:
    one requiring maximum excavation and pilings;
    one requiring minimum excavation, basement at grade level, no pilings for structures

Coal-Fired

Two 500-Mw boilers
2400 psig, 1000 F/1000 F single reheat steam cycle
345 kv output from transformer
Natural draft cooling towers required
Stack height 750 feet
Electrostatic precipitators required for dust collection
Location in southern Illinois
Architectural treatment to yield minimum-price structures
Coal conveyed from mine to plant
Construction site flat and does not require pilings.

TABLE 43

PERFORMANCE OF STEAM POWER STATIONS

	Power System Output, Mw	Power System Efficiency %	Net Station Output Mw (1)	Net Station Efficiency, % (1)
First Generation				
Coal	1012	36.6	1012	36.6
Residual Oil	1022	37.0	1022	37.0
Gasified Coal (Fuel GasII)	1022	37.5	922	28.2
Hydrotreated Oil	1022	37.0	1022	37.0 (2)
Second Generation				
Coal	1012	38.6	1012	38.6
Residual Oil	1022	39.0	1022	39.0
Gasified Coal (Fuel Gas III)	1022	39.6	948	34.0
Hydrotreated Oil	1022	39.0	1022	39.0 (2)
Gasified Oil (Fuel Gas V)	1022	39.6	1005	37.0
Third Generation				
Coal	1012	39.6	1012	39.6
Residual Oil	1022	40.0	1022	40.0
Gasified Coal (Fuel GasIII)	1022	40.7	950	35.0
Gasified Oil (Fuel Gas V)	1022	40.7	1005	38.0

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(1) At 70% load factor

(2) Neglects losses during hydrotreating

TABLE 44

BREAKDOWN OF FPC ACCOUNT 311 - STRUCTURES AND IMPROVEMENTS

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No. 311	Station Type		
Structures and Improvements	Coal-Fired(1)	Oil-Fired(2)	Oil-Fired(3)
Site Improvements			
Test Borings	$ 38,000	$ 50,000	$ 40,000
Clear and Grub	54,000	54,000	54,000
Strip Top Soil	40,000	40,000	40,000
Site Excavation	150,000	450,000	225,000
Fine Grading	39,000	39,000	39,000
Landscaping	100,000	200,000	150,000
Fresh Water Supply	21,000	21,000	21,000
Fire Protection	80,000	80,000	80,000
Sewage Disposal and Drainage	110,000	160,000	130,000
Roads, Walks, and Parking	88,000	104,000	104,000
Railroad spur tracks	115,000	115,000	115,000
Fence and Gates	20,000	20,000	20,000
Structures			
Administration and Service Building	403,000	452,000	403,000
Power House Structure	9,633,000	7,205,000	7,415,000
Gate House	3,000	5,000	3,000
Building Excavation	188,000	175,000	187,000
Dewatering	220,000	218,000	220,000
Piling (Building)	--	1,755,000	--
Miscellaneous			
Boiler Feed Pump Foundations on Building Mat	117,000	117,000	117,000
Condensate Tank Foundations	27,000	64,000	27,000
Stack Foundations	286,000	358,000	225,000
Fuel Oil Tank Farm with Dikes	--	2,528,000	272,000
Draft Plant Foundations	81,000	129,000	81,000
Fuel Oil Unloading Facility	--	610,000	200,000
Misc. Pump Foundations, etc.	310,000	360,000	270,000
Cooling Water Foundations	97,000	310,000	97,000
Circulating Water, Conc. Work in T-G Building	210,000	335,000	210,000
Chlorine House	24,000	30,000	24,000
Hydrogen Storage	11,000	16,000	11,000
Coal Handling Foundations, etc.	600,000	--	--
Total Account No. 311	$13,065,000	$16,000,000	$10,780,000

(1) Minimum site and structural conditions. Plant located in Southern Illinois at mine mouth.
(2) Maximum site and structural conditions. Plant located in Northeast.
(3) Minimum site and structural conditions. Plant located in Northeast.

TABLE 45

BREAKDOWN OF FPC ACCOUNT 312 - BOILER PLANT EQUIPMENT

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No. 312 Boiler Plant Equipment	Station Type Coal-Fired(1)	 Oil-Fired(2)	 Oil-Fired(3)
Steam Generators	$31,700,000	$27,000,000	$27,000,000
Electrostatic Precipitators	1,210,000	--	--
Mechanical Cyclone Dust Separators	--	1,000,000	1,000,000
F.D. Fans with Motor Drive	160,000	479,000	479,000
I.D. Fans with Motor Drive	--	700,000	700,000
Gas Recirculating Fans with Motor	--	181,000	181,000
Boiler Feed Pumps with Spare Rotor	297,000	297,000	297,000
Boiler Feed Pumps Drive Turbines	1,120,000	1,120,000	1,120,000
HP and LP Feedwater Heaters	1,309,000	1,309,000	1,309,000
Deaerating	110,000	110,000	110,000
Auxiliary Boiler	100,000	100,000	100,000
Air Compressors - Serv. and Inst.	150,000	110,000	110,000
Concrete Chimneys	2,250,000	1,700,000	1,700,000
Breeching	1,365,000	990,000	990,000
High Pressure Piping	2,100,000	2,100,000	2,100,000
I.P. and L.P. Piping and Mechanical Installation	10,250,000	10,250,000	10,250,000
Controls and Instrumentation	900,000	875,000	875,000
Water Treatment Equipment	290,000	290,000	290,000
Insulation-Piping and Equipment	1,127,000	1,127,000	1,127,000
Control Valves and Controllers	204,000	204,000	204,000
Misc. Pumps and Motors	240,000	240,000	240,000
Diesel Oil Transfer Pumps	2,000	2,000	2,000
Miscellaneous Tanks	80,000	117,000	117,000
Air Pollution Monitors and Recording Equipment	46,000	46,000	46,000
Powdex Units	270,000	270,000	270,000
Valves	690,000	690,000	690,000
Condensate Tanks	180,000	180,000	180,000
Air Preheat Coils and Heat Exchangers	145,000	145,000	145,000
F.O. Storage and Piping	108,000	2,108,000	2,108,000
Coal Handling System	3,600,000	--	--
Ash Handling System	930,000	--	--
Fire Pumps and Drives	50,000	50,000	50,000
Computer and Accessories	Included in Controls and Instrumentation		
Ammonia and Hydrazine Equipment	Included in Water Treatment		
Sump Pumps	10,000	10,000	10,000
Miscellaneous Tank Heaters	12,000	12,000	12,000
Automatic and Twin Strainers	--	75,000	75,000

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(1) Minimum site and structural conditions. Plant located in Southern Illinois at mine mouth.

(2) Maximum site and structural conditions. Plant located in Northeast.

(3) Minimum site and structural conditions. Plant located in Northeast.

TABLE 45 (Cont'd.)

BREAKDOWN OF FPC ACCOUNT 312 - BOILER PLANT EQUIPMENT (Cont'd.)

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No. 312 Boiler Plant Equipment	Station Type Coal-Fired(1)	 Oil-Fired(2)	 Oil-Fired(3)
Miscellaneous Foundations			
Stack	(4)	(4)	(4)
Condensate Tank	(4)	(4)	(4)
Electrostatic Dust Precipitators	(4)	---	---
Fuel Oil Tanks and Unloading	---	(4)	(4)
Facilities	(4)	(4)	(4)
Mechanical Cyclone Dust Separators	---	(4)	(4)
Coal Handling Equipment	(4)	---	---
Boiler Feed Pumps	(4)	(4)	(4)
Draft Plants	(4)	(4)	(4)
Cooling Tower	(4)	(4)	(4)
Coal Crusher	(4)	---	---
Total - Account 312	$61,005,000	$53,887,000	$53,887,000

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(1) Minimum site and structural conditions. Plant located in Southern Illinois at mine mouth.

(2) Maximum site and structural conditions. Plant located in Northeast.

(3) Minimum site and structural conditions. Plant located in Northeast.

(4) Items included in corresponding categories of Account No. 311.

TABLE 46

BREAKDOWN OF FPC ACCOUNT 314 - TURBOGENERATOR UNITS

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No. 314 Turbogenerator Units	Station Type		
	Coal-Fired(1)	Oil-Fired(2)	Oil-Fired(3)
Turbine Generators	$24,080,000	$24,080,000	$24,080,000
Foundations	570,000	570,000	570,000
Condensers and Tubes	1,846,000	1,846,000	1,846,000
Condenser Vacuum Pumps	135,000	135,000	135,000
Condensate Pumps with Drives	184,000	184,000	184,000
Circulating Water Pumps with Drives	653,000	653,000	653,000
Circulating Water Pump House	330,000	440,000	330,000
Cooling Tower	7,000,000	7,000,000	7,000,000
Traveling Cranes	380,000	380,000	380,000
Chlorination Equipment	30,000	30,000	30,000
Lube Oil Purification Equipment	(4)	(4)	(4)
Lube Oil Fire Protection System	(4)	(4)	(4)
Turbine Oil Pumps	3,000	3,000	3,000
River Water Pump House	95,000	330,000	95,000
Equipment Foundations	(5)	(5)	(5)
Condenser Tube Cleaning Equipment	320,000	320,000	320,000
Circulating Water Piping System	450,000	450,000	450,000
Total Account 314	$36,076,000	$36,421,000	$36,076,000

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(1) Minimum site and structural conditions. Plant located in Southern Illinois at mine mouth

(2) Maximum site and structural conditions. Plant located in Northeast.

(3) Minimum site and structural conditions. Plant located in Northeast.

(4) Included with Mechanical Installation.

(5) Included in appropriate categories of Account No. 311.

TABLE 47

BREAKDOWN OF FPC ACCOUNT 315 - ACCESSORY ELECTRICAL EQUIPMENT

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No. 315 Accessory Electrical Equipment	System Type Coal-Fired(1)	System Type Oil-Fired(2)	System Type Oil-Fired(3)
Station Transformers	$ 243,800	$ 207,000	$ 207,000
Start-up Transformer	189,800	182,300	182,300
Grounding Transformers	34,500	34,500	34,500
Grounding Resistors	2,500	2,500	2,500
23-kv Generator Bus Duct	279,500	279,500	279,500
Current Transformers	43,500	43,500	43,500
Potential Transformers	26,000	26,000	26,000
Surge Protection	12,800	12,800	12,800
13.8-kv Switchgear	500,300	414,000	414,000
13.8-kv Bus Duct	126,500	126,500	126,500
13.8-kv - 480 v Transformers	112,500	112,500	112,500
480-v Power Switch Gear	155,300	152,000	152,000
480-v Motor Control Centers	529,000	527,000	527,000
480-v Valve Control Centers	128,000	128,000	128,000
Remote Motor Controls	17,500	17,500	17,500
Duplex Relay Switch Boards	68,000	68,000	68,000
Annunciator Panels	16,500	16,500	16,500
Control Consoles	34,500	34,500	34,500
Boiler Control Panels	6,000	6,000	6,000
Turbine Control Panels	6,000	6,000	6,000
Temperature Detection Panels	15,000	15,000	15,000
Coal Handling Control Panels	27,000	--	--
Fuel Oil Control Panels	--	27,000	27,000
250-v D.C. Switchboards	27,500	27,500	27,500
250-v D.C. Panelboards	3,600	3,600	3,600
250-v D.C. Motor Controls	1,500	1,500	1,500
Station Battery and Rack	53,000	53,000	53,000
Battery Chargers	56,500	56,500	56,500
Static Inverters	62,000	62,000	62,000
Diesel Engine Generating Equipment	365,400	365,400	365,400
Cable Tray	556,000	556,000	556,000
Conduit and Fittings	2,197,000	2,197,000	2,197,000
Underground Ductwork	313,500	256,500	256,500
15-kv Power Cable	1,097,500	926,500	926,500
600-v Power Cable	983,500	997,800	997,800
600-v Instrument Cable	370,800	370,800	370,800
600-v Control Cable	641,500	641,500	641,500
Grounding System	370,500	370,500	370,500
Equipment Connect	400,000	300,000	300,000
Testing	378,300	378,300	378,300
Lighting and Small Power	600,000	600,000	600,000
Total Account 315	$11,052,600	$10,609,000	$10,603,000

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(1) Minimum site and structural conditions. Plant located in Southern Illinois at mine mouth.
(2) Maximum site and structural conditions. Plant located in Northeast.
(3) Minimum site and structural conditions. Plant located in Northeast.

TABLE 48

BREAKDOWN OF FPC ACCOUNT 316 - MISCELLANEOUS POWER PLANT EQUIPMENT

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No. 316 Miscellaneous Power Plant Equipment	Station Type Coal-Fired(1)	Oil-Fired(2)	Oil-Fired(3)
Laboratory and Sampling Equipment	$ 25,000	$ 20,000	$ 20,000
Tools, Shop, Stores, and Work Equipment	125,000	125,000	125,000
Lockers	3,000	3,000	3,000
Emergency Equipment	10,000	10,000	10,000
Miscellaneous Cranes and Hoists	200,000	200,000	200,000
Portable Fire Extinguishing Equipment	30,000	20,000	20,000
Communications Equipment	40,000	50,000	50,000
Lunch Room Equipment	20,000	20,000	20,000
Office Furniture and Machines	15,000	15,000	15,000
Vacuum Cleaning System	50,000	--	--
Total Account 316	$518,000	$463,000	$463,000

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(1) Minimum site and structural conditions. Plant located in southern Illinois at mine mouth.

(2) Maximum site and structural conditions. Plant located in Northeast.

(3) Minimum site and structural conditions. Plant located in Northeast.

TABLE 49

BREAKDOWN OF FPC ACCOUNT 353 - STATION EQUIPMENT

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No. 353 Station Equipment	Station Type		
	Coal-Fired(1)	Oil-Fired(2)	Oil-Fired(3)
Main Power Transformer	$1,483,000	$1,474,000	$1,474,000
Fog Spray-Fire Protection for all Transformers	35,000	35,000	35,000
Transformer Foundations	157,000	215,000	157,000
Fire Wall Barriers	45,000	45,000	45,000
Total Account 353	$1,720,000	$1,769,000	$1,711,000

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(1) Minimum site and structural conditions. Plant located in southern Illinois at mine mouth.

(2) Maximum site and structural conditions. Plant located in Northeast.

(3) Minimum site and structural conditions. Plant located in Northeast.

TABLE 50

CAPITAL COST SUMMARY FOR STEAM POWER SYSTEMS

First Generation - Two 500-Mw (Nominal) Generating Units

FPC Account No.		Station Type		
		Coal-Fired(1)	Oil-Fired - A(2)	Oil-Fired - B(3)
310	Land and Land Rights	$ 30,000	$ 225,000	$ 225,000
311	Structures & Improvements	13,065,000	16,000,000	10,780,000
312	Boiler Plant Equipment	61,005,000	53,887,000	53,887,000
·314	Turbogenerator Units	36,076,000	36,421,000	36,076,000
315	Accessory Electric Equipment	11,052,000	10,603,000	10,603,000
316	Misc. Power Plant Equipment	518,000	463,000	463,000
353	Station Equipment	1,720,000	1,769,000	1,711,000
		$123,466,000	$119,368,000	$113,745,000
	Other Expenses	1,250,000	1,250,000	1,250,000
		$124,716,000	$120,618,000	$114,995,000
	Engineering, Design, Construction Supervision, and Contingency	12,236,000	11,831,000	11,550,000
		$136,952,000	$132,449,000	$126,545,000
	Escalation	20,200,000	19,536,000	18,665,000
		$157,152,000	$151,985,000	$145,210,000
	Interest During Construction	19,801,000	19,150,000	18,296,000
	Total Estimated Station Cost	$176,953,000	$171,135,000	$163,506,000

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(1) Minimum site and structural conditions. Plant located in southern Illinois at mine mouth. Coal conveyor from mine not included in station cost.

(2) Maximum site and structural conditions. Plant located in Northeast.

(3) Minimum site and structural conditions. Plant located in Northeast.

TABLE 51

CAPITAL COST SUMMARY FOR STEAM POWER SYSTEMS

Second and Third Generation - One 1000-Mw (Nominal) Generating Unit

FPC Account No.	Description	Station Type		
		Coal-Fired(1)	Oil-Fired(2)	Oil-Fired(3)
310	Land and Land Rights	$ 30,000	$ 225,000	$ 225,000
311	Structures and Improvements	9,107,500	12,318,100	7,804,500
312	Boiler Plant Equipment	55,492,100	50,054,500	50,054,500
314	Turbine-Generator Units	34,612,100	34,763,000	34,427,500
315	Accessory Electric Equipment	10,138,480	9,730,000	9,730,000
316	Misc. Power Plant Equipment	463,000	518,000	518,000
353	Station Equipment	1,571,700	1,621,600	1,621,600
	Total	$111,414,880	$109,230,200	$104,381,100
	Other Expenses	1,250,000	1,250,000	1,250,000
		$112,664,880	$110,480,200	$105,631,100
	Engineering, Design, Construction Supervision and Contengency	13,236,000	12,831,000	12,550,000
		$125,900,880	$123,311,200	$118,181,100
	Escalation	18,570,380	18,188,402	17,431,712
		$144,471,260	$141,499,602	$135,612,812
	Interest	18,203,220	17,828,874	17,087,214
	Total Estimated Cost	$162,674,480	$159,328,476	$152,700,026

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(1) Minimum site and construction conditions. Plant located in southern Illinois at mine mouth.

(2) Maximum site and construction conditions. Plant located in Northeast.

(3) Minimum site and construction conditions. Plant located in Northeast.

TABLE 52

INCREMENTAL COST TO PLANTS FOR STANDBY OIL BURNING CAPABILITY

Item	Estimated Cost
Fuel Oil Tanks (250,000 bbl, 1 week supply)	$ 500,000
Foundations, etc.	100,000
Boiler Incremental Cost	300,000
	900,000
Escalation	132,750
	1,032,750
Interest during Construction	130,000
	1,162,750
Oil Inventory (250,000 bbl @ $2.25/bbl)	562,500
Total Incremental Cost	$ 1,725,250

TABLE 53

CAPITAL COST SUMMARY FOR STEAM POWER SYSTEMS

First Generation - Two 500-Mw (Nominal) Generating Units
Estimates are for either Gasified Coal or Gasified Oil Power Systems

FPC Acct. No.	Description	Power System Type I(1)	 II(2)
310	Land and Land Rights	$ 225,000	$ 225,000
311	Structures and Improvements	12,800,000	10,380,000
312	Boiler Plant Equipment	42,600,000	42,600,000
314	Turbine-Generator Units	36,421,000	36,076,000
315	Accessory Electric Equipment	10,603,000	10,603,000
316	Misc. Power Plant Equipment	463,000	463,000
353	Station Equipment	1,769,000	1,711,000
		104,881,000	102,058,000
	Other Expenses	1,250,000	1,250,000
		106,131,000	103,308,000
	Engineering, Design, Construction Supervision, and Contingency	11,331,000	11,050,000
		117,462,000	114,358,000
	Escalation	17,325,600	16,867,800
		134,787,600	131,225,800
	Interest	16,983,000	16,534,000
	Subtotal	$151,770,600	$147,759,800
	Incremental Cost for Standby Oil	1,725,250	1,725,250
	Total Estimated Cost	$153,495,850	$ 149,485,050

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(1) Maximum site and structural conditions.
(2) Minimum site and structural conditions.

TABLE 54

CAPITAL COST SUMMARY FOR STEAM POWER SYSTEMS

Second and Third Generation - One 1000-Mw (Nominal) Generating Unit
Estimates are for Either Gasified Coal or Gasified Oil Power System
with Minimum Site and Structural Conditions

FPC Account No.	Description	Power System Cost Estimates I(1)	II(2)	III(3)
310	Land and Land Rights	$ 30,000	$ 225,000	$ 225,000
311	Structures and Improvements	7,582,000	7,180,180	7,332,000
312	Boiler Plant Equipment	39,200,000	39,200,000	39,200,000
314	Turbine-Generator Units	34,612,000	34,763,000	34,427,500
315	Accessory Electric Equipment	10,138,480	9,730,000	9,730,000
316	Misc. Power Plant Equipment	463,000	518,000	518,000
353	Station Equipment	1,571,700	1,621,600	1,621,600
	Total	$ 93,597,180	$ 93,237,780	$ 93,054,100
	Other Expenses	1,250,000	1,250,000	1,250,000
		$ 94,847,180	$ 94,487,780	$ 94,304,100
	Engineering, Design, Construction Supervision, and Contingency	12,436,000	12,031,000	11,750,000
		$107,283,180	$106,518,780	$106,054,100
	Escalation	15,824,269	15,711,520	15,642,980
		$123,107,449	$122,230,300	$121,697,080
	Interest	15,511,538	15,401,018	15,333,832
	Subtotal	$138,618,987	$137,631,318	$137,030,912
	Average Cost(4)		$137,760,406	
	Incremental Cost for Standby Oil(5)		1,725,250	
	Total Estimated Cost		$139,485,656	

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(1) Adapted from estimates for coal station with minimum site and construction conditions.
(2) Adapted from estimates for oil station with maximum site and construction conditions.
(3) Adapted from estimates for oil station with minimum site and construction conditions.
(4) Averaged to reflect estimate of same plant independent of location.
(5) Includes all equipment, design engineering, contingencies, escalation, interest, etc.

TABLE 55

COST SUMMARY FOR FIRST-GENERATION STEAM POWER STATIONS

Fuel	Coal	Residual Oil(1)	Residual Oil(2)	Gasified Coal (Fuel Gas II)	Hydrotreated Oil(2)
Net Power Station Output, Mw	1012	1022	1022	922	1022
Capital Investment, Thousand Dollars					
Power System	176,953	171,135	163,506	149,485	163,506
Fuel Processing System	-	-	-	51,354	-
Total Station	176,953	171,135	163,506	200,839	163,506
Net Capital Cost, $/kw	174.8	167.5	160.0	217.9	160.0
Annual Owning and Operating Cost, mills/kwhr					
Capital Charges(3)	3.990	3.824	3.653	4.975	3.653
Operation, Supplies, and Maintenance					
Power System	.372	.360	.360	.366	.360
Fuel Processing System(4)	-	-	-	.894	-
Fuel(5)	1.865	2.765	2.765	2.413	4.074(6)
Busbar Power Cost, mills/kwhr	6.227	6.949	6.778	8.648	8.087

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(1) Maximum site and structural conditions
(2) Minimum site and structural conditions
(3) Capital charges @ 14% per annum @ 70% load factor
(4) Includes credit for sulfur
(5) Coal @ 20¢/million Btu, Residual Oil @ 30¢/million Btu
(6) Includes owning and operating costs for hydrotreating facility

TABLE 56

COST SUMMARY FOR SECOND-GENERATION STEAM POWER STATIONS

Fuel	Coal	Residual Oil(1)	Residual Oil(2)	Gasified Coal (Fuel Gas III)	Gasified Oil (Fuel Gas V)
Net Power Station Output, Mw	1012	1022	1022	948	1005
Capital Investment, Thousand Dollars					
Power System	162,674	159,328	152,700	139,486	139,486
Fuel Processing System	-	-	-	33,117	25,626
Total Station	162,674	159,328	152,700	172,603	165,112
Net Capital Cost, $/kw	160.7	155.9	149.4	182.0	164.4
Annual Owning and Operating Cost, mills/kwhr					
Capital Charges(3)	3.668	3.560	3.412	4.154	3.753
Operation, Supplies, and Maintenance					
Power System	0.322	0.312	0.312	0.315	0.315
Fuel Processing System(4)	-	-	-	0.396	0.626
Fuel(5)	1.768	2.625	2.625	2.002	2.760
Busbar Power Cost, mills/kwhr	5.758	6.497	6.349	6.867	7.454

---

(1) Maximum site and structural conditions
(2) Minimum site and structural conditions
(3) Capital charges @14% per annum @ 70% load factor
(4) Includes credit for sulfur
(5) Coal @ 20¢/million Btu, Residual Oil @ 30¢/million Btu

TABLE 57

COST SUMMARY FOR THIRD-GENERATION STEAM POWER STATIONS

Fuel	Coal	Residual Oil(1)	Residual Oil(2)	Gasified Coal (Fuel Gas III)	Gasified Oil (Fuel Gas V)
Net Power Station Output, Mw	1012	1022	1022	950	1005
Capital Investment, Thousand Dollars					
Power System	162,674	159,328	152,700	139,486	139,486
Fuel Processing System	-	-	-	32,449	24,321
Total Station	162,674	159,328	152,700	171,935	163,807
Net Capital Cost, $/kw	160.7	155.9	149.4	180.9	163.0
Annual Owning and Operating Cost, mills/kwhr					
Capital Charges(3)	3.668	3.560	3.412	4.131	3.721
Operation, Supplies, and Maintenance					
Power System	0.322	0.312	0.312	0.314	0.315
Fuel Processing System(4)	-	-	-	0.388	0.588
Fuel(5)	1.724	2.560	2.560	1.948	2.690
Busbar Power Cost, mills/kwhr	5.714	6.432	6.284	6.781	7.314

---

(1) Maximum site and structural conditions
(2) Minimum site and structural conditions
(3) Capital charges @ 14% per annum @ 70% load factor
(4) Includes credit for sulfur
(5) Coal @ 20¢/million Btu, Residual Oil @ 30¢/million Btu

TABLE 58

PROJECTED DESIGN TECHNOLOGY FOR BASE-LOAD GAS TURBINE SYSTEMS

Parameter	Time Period		
	First Generation	Second Generation	Third Generation
Turbine Inlet Gas Temperature, F	1600 to 2200(1) 2200 to 2400(2)	2000 to 2400(1) 2600 to 2800(2)	2400 to 2800(1) 3000 to 3100(2)
Compressor Pressure Ratio	8 to 28	8 to 36	8 to 36
Compressor Polytropic Efficiency, %	89	92	93
Turbine Nominal Adiabatic Efficiency, %	90	92	93
Regenerator Air Side Effectiveness, %	60 to 90	60 to 90	60 to 90
Regenerator Total Pressure Drop, %	4 to 8	4 to 8	4 to 8
Turbine Cooling Technique	Impingement-Convection	Impingement-Convection	Impingement- Convection Transpiration

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(1) Compressor Bleed Air Uncooled Prior to Turbine Cooling.

(2) Compressor Bleed Air Precooled from 125 to 250 F Prior to Turbine Cooling.

TABLE 59

BASE-LOAD GAS TURBINE DESIGN PARAMETERS

Component	Parameter	Time Period First Generation	 Second Generation	 Third Generation
Inlet	Stack pressure loss, in. H_2O		← 4.0 →	
Low Compressor	Inlet pressure loss ($\Delta P/P$), %		← 0.76 →	
	Stage efficiency (polytropic), %	89	92	93
	Corrected airflow per unit flow area, lb/sec/ft^2		← 30 to 38 —	
	Corrected tip speed ($V_T/\sqrt{\theta}$), ft/sec	1000	1100	1200
	First stage hub/tip ratio*		← 0.55 to 0.70 →	
	Maximum aspect ratio		← 3.0 →	
High Compressor	Stage efficiency (polytropic), %	89	92	93
	Corrected airflow per unit flow area, lb/sec/ft^2		← 30 to 35 →	
	Maximum corrected tip speed, ($V_T/\sqrt{\theta}$), ft/sec	1000	1100	1200
	Maximum exit hub/tip ratio*		← 0.93 →	
	Minimum aspect ratio		← 1.0 —	
Combustor	Combustion efficiency, %		← 100 —	
	Total pressure loss ($\Delta P/P$), %		— 3.0 →	
High Turbine	Maximum stage work, Btu/lb*	155	175	200
	Nominal stage efficiency (polytropic), %	88	91	92
	Hub/tip ratio		— 0.825 to 0.875 —	
	Blade height/axial width$^{(1)}$*		← 1.0 to 2.5 —	
	Taper ratio	0.8	0.75	0.667
	Minimum blade axial width$^{(1)}$, in.		— 1.0 →	
	Minimum vane axial width$^{(1)}$, in.		← 1.5 →	
	Allowable blade metal temperature, F	1370	1570	1740 to 1770
Low Turbine	Nominal stage efficiency (polytropic), %	88	91	92
	Last stage hub/tip ratio*		← 0.75 to 0.50 →	
	Blade height/axial width$^{(1)}$*		← 2.0 to 5.5 →	
	Taper ratio	0.8	0.75	0.667
	Minimum blade axial width$^{(1)}$, in.		— 1.5 →	
	Minimum vane axial width$^{(1)}$, in.		← 1.5 →	
	Exit diffuser losses ($\Delta P/P$), %		← 1.5 →	
Power Turbine	Total efficiency (adiabatic), %	90	92	93
	Exit velocity, ft/sec		← 400 to 850 →	
	Last stage hub/tip ratio*		← 0.50 to 0.58 →	
	Rotational speed, rpm		← 1800 →	
	Minimum blade axial width$^{(1)}$, in.		← 1.5 →	
	Blade height/axial width$^{(1)}$*		← 2.0 to 5.5 →	
	Taper ratio	0.8	0.75	0.667
	Exit diffuser losses ($\Delta P/P$), %		← 1.5 →	
	Maximum Allowable Blade Root Stress, psi	40,000	40,000	47,000

* These parameters represent design constraints that must be satisfied.

(1) Defined at mean blade or vane height.

TABLE 60

BASE-LOAD GAS TURBINE DESIGNS

	First Generation	Second Generation	Third Generation
Number of Exhaust Ends	1	1	1
Output/unit (Mw)	197	331	506
Fuel	Fuel Gas III	Fuel Gas II	Fuel Gas II
Compressor Pressure Ratio	20	32	34
Station Efficiency[1], (%)	35.2	41.5	43.6
Turbine Inlet Temperature (F)	2200	2800	3100
Compressor Airflow (lb/sec)	1029	968.2	1094
Exhaust Gas Flow (lb/sec)	1225	1250	1500
Exhaust Gas Temp. (F)	951	1105	1256
% Bleed for Turbine Cooling	5.72	10.8	10.7
Gas Generator Length (ft)	20.2	15.73	17.73
Inlet Dia. (ft)	9.24	8.38	8.97
Combustor Max. Dia. (ft)	6.56	5.98	6.30
Power Turbine Length[2] (ft)	10.35	10.19	10.97
Power Turbine Exit Dia. (ft)	10.1	10.2	10.93
Overall Engine Length (ft)	30.55	25.92	28.69
Exit Axial Velocity(ft/sec)	750	750	800
Turbine Cooling Air Temp. (F)	200	200	200
Compressor Exit Temp. (F)	922.4	1075	1084

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(1) Includes 3% auxiliary, mechanical, and generator losses

(2) Based on one exhaust end

TABLE 61

ITEMIZED COSTS FOR GAS TURBINE POWER STATION

FPC Account No. 341 - Structures and Improvements

Structures and Improvements	First Generation	Second Generation	Third Generation
Site Improvements			
Site Grading	$ 25,000	$ 25,000	$ 25,000
Building Excavation	13,000	7,500	6,000
Borings	6,400	5,500	5,000
Landscaping	25,300	22,000	20,000
Fresh Water Supply	12,700	12,000	12,000
Fire Protection	101,200	100,000	90,000
Drain and Sewage Disposal	19,000	19,000	19,000
Flagpole	5,000	5,000	5,000
Guard House	7,600	7,600	7,600
Railroad	50,600	50,600	50,600
Roads and Parking Lots	20,900	20,900	20,900
Fencing	41,800	41,800	41,800
Switchyard	10,700	10,700	10,700
Structures			
Administration Building	407,500	403,500	431,000
T.G. Building	3,916,000	2,669,000	2,414,900
Tank Farm	855,300	855,300	855,300
Fuel-Oil Pump House	35,800	35,800	35,800
Gas Meter Area	2,700	2,700	2,700
Total	$5,556,500	$4,293,900	$4,053,300

TABLE 62

ITEMIZED COSTS FOR GAS TURBINE POWER STATIONS

FPC Account No. 343 - Prime Movers

	First Generation	Second Generation	Third Generation
Gas Turbines	$26,500,000	$19,350,000	$21,000,000
Start-Up Motors	40,000	22,500	16,000
Torque Converter	375,000	225,000	150,000
Lube Oil Purif. & Storage	88,000	72,000	60,000
Lube Oil Fire Protect.	100,000	60,000	50,000
Turb. Air Precoolers	90,000	96,000	96,000
Air Comp. Serv. & Inst.	50,000	50,000	50,000
Breeching Incl. Lining, Silencers & Insulation	3,160,000	2,025,000	2,554,000
Expansion Joints	150,000	90,000	60,000
Inlet Filter Screens	150,000	100,000	75,000
Turb. Encls. Air Cooler	100,000	60,000	40,000
Emerg. Cooling Water Tank, Pump & Piping	10,000	10,000	10,000
F.O. Heaters & Pumps	22,000	22,000	22,000
Misc. Pumps & Tanks	25,000	15,000	10,000
Cont. Boards, Instruments and Controls	250,000	150,000	100,000
Computer	250,000	200,000	200,000
Piping	1,000,000	900,000	800,000
Insulation	150,000	135,000	120,000
Total	$32,510,000	$23,582,500	$25,413,000

TABLE 63

ITEMIZED COSTS FOR GAS TURBINE POWER STATIONS

FPC Account No. 344 - Electric Generators

	First Generation	Second Generation	Third Generation
Generators	$ 9,625,000	$ 9,570,000	$ 9,460,000
H_2 Seal Oil Coolers	50,000	30,000	20,000
Total	$ 9,675,000	$ 9,600,000	$ 9,480,000

TABLE 64

ITEMIZED COSTS FOR GAS TURBINE POWER STATIONS

FPC Account No. 345 - Accessory Electrical Equipment

	First Generation	Second Generation	Third Generation
Auxiliary Transformers	$ 48,500	$ 29,100	$ 19,400
Start-Up Transformers	35,000	86,200	86,200
8000 A Isolated Phase Bus	1,122,800	--	--
1200A Isolated Phase Bus	103,500	62,100	41,400
Potential XFMRS.	65,000	39,000	26,000
Surge Protection	32,000	19,200	12,800
480-v Power Switchgear	114,275	77,325	58,775
480-v Motor Cont. Cntrs.	46,615	33,235	27,655
Remote Motor Controls	3,500	2,625	1,750
Duplex Relay Switchboard	68,000	68,000	68,000
Annunciator Panel	16,500	16,500	16,500
Control Console	34,500	34,500	34,500
Turbine Control Panel	6,000	6,000	6,000
Temp. Detection Panel	15,000	15,000	15,000
Equipment Connect	1,800,000	1,800,000	1,800,000
Testing	378,300	378,300	378,300
250-v DC Switchboard	27,500	27,500	27,500
250-v DC Panelboard	3,600	3,600	3,600
Station Battery and Rack	53,000	53,000	53,000
Battery Chargers	56,500	56,500	56,500
Cable Tray	82,000	82,000	82,000
600-v Instrument Cable	80,000	60,400	80,000
600-v Control Cable	160,000	122,000	160,000
Grounding System	370,500	370,500	370,500
480-v Valve Control Center	25,600	25,600	25,600
Conduit-Fittings	61,205	48,200	36,700
600-v Power Cable	31,330	23,935	20,210
1000-v Power Cable	43,545	35,125	26,995
12,000A Isolated Phase Bus	--	1,073,900	--
18,000A Isolated Phase Bus	--	--	935,300
Total	$4,884,270	$4,649,345	$4,470,685

TABLE 65

ITEMIZED COSTS FOR GAS TURBINE POWER STATIONS

FPC Account No. 346 - Miscellaneous Power Plant Equipment

	First Generation	Second Generation	Third Generation
Misc. P.P. Equipment			
Lab. & Samp. Equipment	$ 10,000	$ 10,000	$ 10,000
Tools, Shop, Stores and Work Equipment	75,000	75,000	75,000
Lockers	3,000	3,000	3,000
Emergency Equipment	10,000	10,000	10,000
Misc. Cranes and Hoists	15,000	15,000	15,000
Portable Fire Extinguishers	20,000	20,000	20,000
Communications Equipment	50,000	50,000	50,000
Lunch Room Equipment	20,000	20,000	20,000
Office Furniture & Machines	15,000	15,000	15,000
Total	$218,000	$218,000	$218,000

TABLE 66

ITEMIZED COSTS FOR GAS TURBINE POWER STATIONS

FPC Account No. 353 - Station Equipment

	First Generation	Second Generation	Third Generation
345-kv Main Oil C/B	$ 402,000	$ --	$ --
132-kv Outdoor Swgr.	398,300	--	--
235 MVA Transformer	1,765,000	--	--
470 MVA Auto-XFMR.	639,000	--	--
705 MVA Auto-XFMR.	829,000		
388 MVA Transformer		1,719,000	--
400 MVA Transformer	--	--	--
588 MVA Transformer			1,376,000
Total	$4,033,300	$1,719,000	$1,376,000

TABLE 67

CAPITAL COST SUMMARY FOR ADVANCED-CYCLE GAS TURBINE POWER STATIONS

1000-Mw Nominal Output

Federal Power Commission Acct. No.	Description	First Generation	Second Generation	Third Generation
		(Capital Costs, Thousand Dollars)		
340	Land and Land Rights	225	225	225
341	Structures and Improvements	5,557	4,294	4,053
343	Prime Movers	32,510	23,583	25,413
344	Electric Generators	9,675	9,600	9,480
345	Accessory Electrical Equipment	4,884	4,649	4,471
346	Miscellaneous Power Plant Equipment	218	218	218
353	Station Equipment	4,033	1,719	1,376
	Other Expenses	1,250	1,250	1,250
Total Direct Cost		58,352	45,538	46,486
	Engineering, Design, Construction Supervision	6,398	5,467	5,224
	Escalation During Construction	6,104	4,611	4,699
	Interest During Construction	6,600	5,228	5,303
Total Installed Power Plant Cost		77,514	60,844	61,712

TABLE 68

PROJECTED DESIGN TECHNOLOGY FOR BASE-LOAD GAS TURBINE SYSTEMS

Parameter	Time Period First Generation (1970's)	 Second Generation (1980's)	 Third Generation (1990's)
Turbine Inlet Gas Temperature, F	1600 to 2400	2000 to 2800	2400 to 3100
Compressor Pressure Ratio	8 to 28	8 to 36	8 to 36
Compressor Polytropic Efficiency, %	89	92	93
Turbine Nominal Adiabatic Efficiency, %	90	92	93
Turbine Cooling Technique	Impingement-Convection	Impingement-Convection	1. Impingement-Convection 2. Transpiration
Turbine Cooling Air Temperature, F	1. Same as Compressor Discharge Air Temperature 2. Precooled to 125-to-250 F		

TABLE 69

GAS-SIDE PRESSURE DROPS FOR EXHAUST-FIRED AND SUPERCHARGED BOILER SECTIONS

Component	Pressure Drop, %(1)
Furnace	1.50
Reheater	1.50
Superheater	1.50
Economizer	2.00
Stack Gas Cooler	2.00

---

(1) Based on gas pressure at inlet to boiler section.

TABLE 70

STEAM CONDITIONS FOR WASTE-HEAT STEAM CYCLES

Steam Temperature, F	Steam Pressure, psig
1000/1000	3500
1000/770	3500
1000/1000	2400
1000/705	2400
1000/1000	1800
1000	1800
950/950	1800
950	1800
900	1450
900	1250
825	850
750	600
750	400
650	400

TABLE 71

PERFORMANCE COMPARISON FOR WASTE-HEAT STEAM SYSTEMS

Gas temperature into boiler = 1226 F
No extractions for feedwater heating
Condenser pressure = 2.0 in. Hg

Steam Cycle Conditions			Net Steam Rate(1)	Net Heat Rate	Steam/Gas Flow Ratio(2)	Stack Temperature(3)	Steam Output
psig /	F /	F	lb/kwhr	Btu/kwhr	lb steam/lb gas	F	kw/lb gas/sec
3500	1000	1000	5.491	8613	0.1398	413.7	91.6
3500	1000	770(4)	6.435	9262	0.1607	369.5	89.9
2400	1000	1000	5.545	8802	0.1463	366.2	94.9
2400	1000	705(4)	6.614	9420	0.1742	306.8	94.8
1800	1000	1000	5.691	9018	0.1478	358.5	93.5
1800	1000	--	6.897	9676	0.1793	294.5	93.6
1250	900	--	7.482	10201	0.1888	272.6	90.8

(1) Adjusted for boiler feed pump power and 1.1% generator and mechanical losses , steam leakage neglected;
(2) Based on pinch temperature of 50 F, mean gas specific heat of 0.27 Btu/lb F
(3) Based on feedwater temperature adjusted for enthalpy rise through boiler feed pump.
(4) Minimum allowable reheat temperature, corresponds to 13% moisture in steam when expaned to 2.0 in. Hg.

TABLE 72

COMPARATIVE COSTS OF SECONDARY STEAM SYSTEMS

	Specific Cost, $/kw Case Number		
	I	II	III
Secondary turbine-generator cost	55	--	39
Incremental main turbine cost	--	37	--
Secondary condenser cost	4	--	--
Incremental main condenser cost	--	4	4
Secondary steam boiler cost	119	119	119
Incremental main turbine building cost	11	11	--
Subtotal	189	171	162
Other construction costs	100	100	100
Total Secondary Steam System Cost	289	271	262
Differential secondary steam system cost	27	9	base

TABLE 73

HEAT TRANSFER SURFACE COST

Tube or Pipe OD, in.	Tube Cost $/ft^2 1.675	Tube Cost $/ft^2 2.000	Tube Cost $/ft^2 2.375	Tube Cost $/ft^2 2.750	Pipe Cost $/ft^2 2.375
Economizer (carbon steel)	2.00	1.93	1.98	2.02	1.70
Vaporizer (carbon steel)	7.52	7.20	7.41	7.57	4.92
Primary Superheater (P-11 or T-11)	10.70	10.71	11.68	12.11	21.70
Reheater (P-22 or T-22)	9.17	9.28	8.78	9.29	24.29
Secondary Superheater (P-22 or T-22)	13.85	15.41	16.30	16.71	24.29

TABLE 74

PRELIMINARY PERFORMANCE AND HEAT BALANCE DATA FOR SECOND-GENERATION WASTE-HEAT COGAS SYSTEMS

Turbine Inlet Temperature - 2800 F
Compressor Pressure Ratio - 16
Fuel Gas I Delivered at 150 F and Burner Pressure

	Gas Turbine Back Pressure - in. H_2O Gauge				
	8.3	20	50	100	200
Gas Turbine Output (kw)	684,092	678,409	664,387	642,260	602,149
Steam Turbine Output (kw)	323,995	329,769	344,015	366,404	407,246
Gas Flow (lb/hr)	9,108,840	9,137,630	9,219,220	9,348,800	9,612,750
Steam Flow (lb/hr)	1,814,330	1,847,220	1,927,890	2,053,280	2,282,040
Overall Plant Heat Rate (Btu/kw)	6,188	6,209	6,263	6,351	6,530
Economizer Duty (million Btu/hr)	1,226	1,248	1,302	1,387	1,541
Gas Inlet Temp. (F)	757	759	764	773	793
Gas Outlet Temp (F)	243	237	224	205	179
Water Inlet Temp. (F)	110	110	110	110	110
Water Outlet Temp. (F)	681	681	681	681	681
Vaporizer Duty (million Btu/hr)	552	562	587	625	694
Gas Inlet Temp. (F)	977	982	995	1,015	1,054
Gas Outlet Temp. (F)	757	759	764	773	793
Water Inlet Temp. (F)	681	681	681	681	581
Pressure Drum (psia)	2,730	2,730	2,730	2,730	2,730
Prim. Superheater Duty (million Btu/hr)	490	499	521	555	617
Gas Inlet Temp. (F)	1,168	1,176	1,195	1,225	1,279
Gas Outlet Temp. (F)	977	982	995	1,015	1,054
Steam Inlet Temp. (F)	681	681	681	681	681
Steam Outlet Temp. (F)	841	841	841	841	841
Reheater Duty (million Btu/hr)	351	358	374	398	443
Gas Inlet Temp. (F)	1,303	1,313	1,336	1,372	1,437
Gas Outlet Temp.(F)	1,168	1,176	1,195	1,225	1,279
Steam Inlet Temp. (F)	655	655	655	655	655
Steam Outlet Temp. (F)	1,000	1,000	1,000	1,000	1,000
Second Superheater Duty (million Btu/hr)	226	230	241	256	285
Gas Inlet Temp. (F)	1,388	1,399	1,425	1,465	1,537
Gas Outlet Temp. (F)	1,303	1,313	1,336	1,372	1,437
Steam Inlet Temp. (F)	841	841	841	841	841
Steam Outlet Temp. (F)	1,000	1,000	1,000	1,000	1,000

TABLE 75

COST DATA FOR SECOND-GENERATION WASTE-HEAT BOILERS

Turbine Inlet Temperature = 2800 F
Compressor Pressure Ratio = 16
Boiler Pipe OD = 2.375 in.

Fuel Cost = 40¢/million Btu
Capital Charges = 14%

Description	Gas Turbine Back Pressure in. H_2O Gauge				
	8.3	20	50	100	200
Heat Recovery Boiler					
Economizer Surface (ft^2)	906,498	736,301	722,666	656,195	688,457
Vaporizer (ft^2)	242,580	208,259	164,965	142,694	111,469
Primary Superheater (ft^2)	125,347	97,965	75,171	67,225	53,720
Reheater (ft^2)	82,818	67,654	47,844	41,857	36,858
Secondary Superheater (ft^2)	43,983	40,738	27,158	23,503	19,772
Total Heat Transfer Surface (ft^2)	1,401,226	1,150,917	1,037,804	931,474	910,276
Cost of Heat Transfer Surface ($)	8,443,73	6,961,170	5,421,056	4,798,483	4,191,628
Cost of Enclosure ($)	1,410,10	1,322,622	1,355,264	1,679,469	2,305,395
Field Erection ($)	2,956,15	2,485,13	2,032,896	1,943,386	1,949,107
Controls, Undistr. & Escalation ($)	3,074,3	2,584,5	2,114,212	2,021,121	2,027,071
Total Boiler Cost ($)	15,884,3	13,353,47	10,923,428	10,442,459	10,473,201
Diff. Boiler Cost ($)	4,960,963	2,430,045	BASE	- 480,969	- 450,227
Turbine-Generator Plant					
Gross Generation (kw)	1,008,087	1,008,178	1,008,402	1,008,754	1,009,495
Gas Turbine Output (kw)	684,092	678,409	664,387	642,260	602,149
Steam System Output (kw)	323,995	329,769	344,015	366,494	407,246
Diff. Output of Gas Turbine (kw)	+ 19,705	+ 14,022	BASE	- 22,127	- 62,238
Diff. Output of Steam System (kw)	- 20,020	- 14,246	BASE	+ 22,479	+ 63,231
Diff. [illegible] Cost ($)	- 710,150	- 505,330	BASE	+ 797,325	+2,242,875
Total Diff. Capital Cost ($)	+4,250,813	+1,924,715	BASE	+ 316,356	+1,792,648
Diff. Annual Fixed Charges ($)	+ 595,114	+ 269,460	BASE	+ 44,290	+ 250,970
Annual Fuel Cost ($)	15,346,984	15,398,642	15,531,496	15,750,257	16,193,954
Diff. Annual Fuel Cost ($)	- 184,512	- 132,854	BASE	+ 218,761	+ 662,458
Total Evaluated Differ. Annual Cost ($)	+ 410,602	+ 136,606	BASE	+ 263,051	+ 913,428
NOTES:					
Overall Plant Heat Rate	6188.30	6209.13	6262.70	6350.91	6529.82
Boiler Cost/kw Stm.	49.03	40.49	31.75	28.49	25.72
Boiler Cost/ft^2 Surface	11.34	11.60	10.52	11.21	11.50

TABLE 76

TUBE SIZE COMPARISON FOR WASTE HEAT BOILERS

Second-Generation COGAS Systems
Gas Turbine Back-pressure = 50 in. H_2O Gauge
Turbine Inlet Temperature = 2800 F
Compressor Pressure Ratio = 16

	Tube O. D. in.				Pipe O.D. in.
	1.625	2.000	2.375	2.750	2.375
Tube Length, ft					
Economizer and Vaporizer	648,834	517,196	443,396	382,469	443,396
Primary Superheater	201,598	154,709	120,899	105,300	120,899
Reheater	127,199	94,949	76,949	64,350	76,949
Secondary Superheater	66,960	54,000	43,680	36,960	43,680
Weights, lb					
Economizer Fins	442,985	437,496	436,837	439,418	436,837
Economizer and Vaporizer	2,381,871	2,668,213	3,056,329	3,380,641	3,307,734
Primary Superheater	740,068	798,142	902,148	1,001,610	901,905
Reheater	311,255	345,424	337,883	380,951	386,283
Secondary Superheater	315,113	394,577	437,237	444,257	394,429
Surface, ft^2					
Economizer and Vaporizer	896,593	883,671	887,631	890,923	887,631
Primary Superheater	85,767	81,006	75,171	75,812	75,171
Reheater	54,114	49,716	47,844	46,330	47,844
Secondary Superheater	28,487	28,274	27,158	26,610	27,158
Total	1,064,961	1,042,667	1,037,804	1,039,675	1,037,804
Cost of Heat Transfer Surface, $					
Economizer and Vaporizer	2,723,708	2,574,614	2,656,669	2,671,459	1,968,603
Primary Superheater	917,280	867,923	877,734	918,216	1,630,941
Reheater	496,080	461,457	420,147	430,502	1,161,945
Secondary Superheater	394,394	435,780	451,651	444,629	659,567
Total	4,531,462	4,339,774	4,406,201	4,464,805	5,421,056
Enclosure Cost, $	1,132,866	1,084,944	1,101,550	1,116,201	1,355,264
Erection Cost, $	1,699,298	1,627,415	1,652,325	1,674,302	2,032,896
Escalation, Contingency Controls, Foundations, & Undistributed Costs, $	1,767,270	1,692,512	1,718,418	1,741,274	2,114,211
Total Boiler Cost, $	9,131,296	8,744,645	8,878,494	8,996,582	10,923,428
Unit Cost, $/kw	26.54	25.42	25.81	26.15	31.75

TABLE 77

SELECTED DATA ON GAS TURBINE PORTIONS OF COGAS POWER SYSTEMS

	First Generation	Second Generation	Third Generation
Number of Gas Turbines	3	2	2
Nominal Output Per Gas Turbine, Mw	193	314	348
Fuel	Fuel Gas III	Fuel Gas II	Fuel Gas II
Number of Spools	1	1	2
Compressor Pressure Ratio	8	12	20
Turbine Inlet Temperature, F	2200	2800	3100
Turbine Cooling Air Temperature, F	200	200	200
Compressor Exit Temperature, F	582	691	861
Exhaust Gas Temperature, F	1297	1514	1485
Compressor Airflow, lb/sec	1110	999	810
Exhaust Gas Flow, lb/sec	1292	1268	1063
Bleed for Turbine Cooling, % of Airflow	4.71	8.54	9.0
Gas Generator Length, ft	20.9	15.3	15.1
Power Turbine Length, ft	12.4	11.4	10.7
Overall Engine Length(1), ft	33.3	26.7	25.8
Inlet Diameter, ft	9.6	8.3	7.7
Combustor Diameter, ft	9.4	7.8	6.6
Power Turbine Exit Diameter, ft	11.3	11.0	10.4
Number of Exhaust Ends	1	1	1
Exit Axial Velocity, ft/sec	850	850	800
Power Turbine Rotational Speed, rpm	1800	1800	1800
Power Turbine Back Pressure, in. H_2O gauge	40	40	40

(1) From compressor inlet stage to power turbine exit stage

TABLE 78

SELECTED DATA ON STEAM PORTIONS OF COGAS POWER SYSTEMS

Steam Turbine	First Generation	Second Generation	Third Generation
Number of Turbines	1	1	1
Nominal Output, Mw	431	381	312
Type	TC-DF	TC-DF	TC-DF
Throttle Temperature, F	1000	1000	1000
Throttle Pressure, psig	2400	2400	2400
Throttle Flow, thousand lb/hr	2413	2136	1750
Reheat Temperature, F	1000	1000	1000
Reheat Pressure, psig	569	569	569
Reheat Flow, thousand lb/hr	2382	2109	1728
Heat Recovery Boiler			
Boiler Inlet Gas Pressure, in. H_2O gauge	40	40	40
Feedwater Temperature to Boiler, F	132	145	159
Secondary Superheater Inlet Gas Temperature, F	1297	1514	1485
Secondary Superheater Duty, Million Btu/hr	301.0	266.5	218.3
Secondary Superheater Surface Area, ft^2	43,361	23,810	20,596
Reheater Inlet Gas Temperature, F	1222	1416	1390
Reheater Duty, million Btu/hr	467.9	414.3	339.3
Reheater Surface Area, ft^2	79,399	44,598	38,423
Primary Superheater Inlet Gas Tempeature, F	1104	1262	1240
Primary Superheater Duty, Million Btu/hr	652.2	577.4	472.9
Primary Superheater Surface Area, ft^2	121,122	71,312	61,718
Vaporizer Inlet Gas Temperature, F	937	1043	1027
Vaporizer Duty, Million Btu/hr	734.1	650.0	532.3
Vaporizer Surface Area, ft^2	261,314	148,033	127,907
Economizer Inlet Gas Temperature, F	745	788	780
Economizer Duty, Million Btu/hr	1577.9	1369.8	1096.5
Economizer Surface Area, ft^2	720,629	796,233	623,188
Final Stack Temperature, F	[illegible]	219	241

TABLE 79

SELECTED PERFORMANCE DATA FOR COGAS POWER SYSTEMS

	First Generation	Second Generation	Third Generation
Gas Turbine Output, Mw	579	628	696
Net Steam System Output, Mw	431	381	312
Power System Auxiliary Power, Mw	10	9	8
Net Power System Output, Mw	1000	1000	1000
Gas Turbine Efficiency, %	27.3	34.2	40.4
Net Steam Cycle Efficiency, %	39.4	39.7	40.1
Net Power System Efficiency, %	47.0	54.5	57.7
Thermal Heat Rejection, Btu/kwhr	2260	1990	1640

TABLE 80

BREAKDOWN OF FPC ACCOUNT 341 - STRUCTURES AND IMPROVEMENTS

FPC Account Number 341 Structure & Improvements	First Generation	Second Generation	Third Generation
Site Improvements			
Site Grading	$ 41,000	$ 41,000	$ 41,000
Building Excavation	24,500	22,000	22,000
Borings	8,900	8,900	8,900
Landscaping	38,000	38,000	38,000
Fresh Water Supply	19,000	19,000	19,000
Fire Protection	126,500	126,500	126,500
Drainage & Sewage Disposal	31,600	31,600	31,600
Flagpole	5,000	5,000	5,000
Guard House	7,600	7,600	7,600
Railroad	94,900	94,900	94,900
Roads & Parking Lots	39,700	39,700	39,700
Fencing	61,500	61,500	61,500
Switchyard	60,300	60,300	60,300
Structures			
Administration Building	526,600	526,600	526,600
Turbine Generator Building	5,284,400	4,537,000	4,537,000
Tank Farm	901,000	901,000	901,000
Fuel-Oil Pump House	35,800	35,800	35,800
Gas Meter Area	2,700	2,700	2,700
Circulating Water System	768,000	768,000	768,000
Stack	415,500	415,500	415,500
Total Account 341	$8,492,400	$7,748,500	$7,742,500

TABLE 81

BREAKDOWN OF FPC ACCOUNT 343 - PRIME MOVERS

FPC Account Number 343 Prime Movers (Gas Turbines)	First Generation	Second Generation	Third Generation
Gas Turbines	$15,700,000	$12,700,000	$13,300,000
Start-Up Motors	24,000	15,000	15,000
Torque Converter	225,000	150,000	150,000
Lube Oil Purification & Storage	72,000	60,000	60,000
Lube Oil Fire Protection	80,000	60,000	64,000
Turbine Air Precoolers	54,000	64,000	64,000
Air Compressors, Service & Installation	100,000	100,000	100,000
Breeching Including Lining, Silencers & Insulation	906,000	720,000	735,000
Expansion Joints	130,000	100,000	100,000
Inlet Filer Screens	90,000	60,000	60,000
Turbine Enclosure Air Cooler	60,000	40,000	40,000
Emergency Cooling Water Tank, Pump & Piping	8,000	8,000	8,000
Fuel Oil Heaters & Pumps	22,000	22,000	22,000
Miscellaneous Pumps & Tanks	15,000	10,000	10,000
Control Boards Instruments & Controls	150,000	100,000	100,000
Computer	200,000	200,000	200,000
Piping	900,000	800,000	800,000
Insulation	135,000	120,000	120,000
Total Account 343	$18,871,000	$15,329,000	$15,948,000

TABLE 82

BREAKDOWN OF FPC ACCOUNT 344 - ELECTRIC GENERATORS

FPC Account Number 344 Electric Generators	First Generation	Second Generation	Third Generation
Electric Generators (for Gas Turbines)	$5,610,000	$5,940,000	$6,600,000
H_2 Seal Oil Coolers	30,000	20,000	20,000
Total Account 344	$5,640,000	$5,960,000	$6,620,000

TABLE 83

BREAKDOWN OF FPC ACCOUNT 312 - BOILER PLANT EQUIPMENT

FPC Account Number 312 Boiler Plant Equipment	First Generation	Second Generation	Third Generation
Waste Heat Boiler	$14,700,000	$ 9,800,000	$ 7,050,000
Boiler Feed Pumps	550,000	474,000	385,000
Boiler Feed Tank and Deaerator	50,000	40,000	31,000
Water Treatment	300,000	240,000	200,000
Condensate Storage Tank	30,000	25,700	24,200
Stack	(Included in Account 341-Stack)		
Process Steam Heat Exchanger	---	---	125,000
Miscellaneous Pumps	100,000	100,000	100,000
Piping	3,200,000	2,900,000	2,600,000
Insulation	280,000	270,000	250,000
Controls (Boiler & Turbine Generator)	300,000	300,000	300,000
Computer (Additive to Gas Turbine)	50,000	50,000	50,000
Total Account 312	$19,560,000	$14,199,700	$11,115,200

TABLE 84

BREAKDOWN OF FPC ACCOUNT 314 - STEAM TURBINE GENERATOR UNITS

FPC Account Number 314 Steam Turbine Generator Units	First Generation	Second Generation	Third Generation
Turbogenerators	$ 9,482,000	$ 8,942,000	$ 8,100,000
Pedestal	(Included in Account 341-(T.G. Bldg)		
Condenser & Tubes	1,120,000	805,000	800,000
Condenser Vacuum Pumps with Motor	90,000	90,000	90,000
Condensate Pumps with Motors	63,000	61,200	42,000
Cooling Towers	4,200,000	4,000,000	2,850,000
Circulating Water Piping	(Included in Account 341 - Circ. Water)		
Circulating Water Valves & Expansion Joints	98,000	90,000	82,000
Circulating Water Pumps	382,500	310,000	266,500
Make-Up Structure Screen & Pumps	300,000	250,000	210,000
Chlorination Equipment	20,000	20,000	20,000
Lube Oil Purification	(Included in Turbogenerator Price)		
Total Account 314	$15,755,500	$14,568,200	$12,460,500

TABLE 85

BREAKDOWN OF FPC ACCOUNT 345 - ACCESSORY ELECTRICAL EQUIPMENT

FPC Account Number 345 Accessory Electrical Equipment	First Generation	Second Generation	Third Generation
Auxiliary Transformers	$ 58,500	$ 39,000	$ 39,000
Start-up Transformers	266,300	297,300	297,300
8000 A Isolated Phase Bus	673,600	432,600	432,600
1200 A Isolated Phase Bus	82,800	82,800	82,800
Potential Transformer	52,000	39,000	39,000
Surge Protection	25,600	19,200	19,200
480 Volt Power Switchgear	136,100	124,300	124,300
480 Volt Motor Control Centers	65,310	58,530	58,530
Remote Motor Controls	5,600	5,250	5,250
Duplex Relay Switchboard	68,000	68,000	68,000
Annunciator Panel	16,500	16,500	16,500
Control Console	34,500	34,500	34,500
Turbine Control Panel	6,000	6,000	6,000
Temperature Detection Panel	15,000	15,000	15,000
Equipment Connections	1,800,000	1,800,000	1,800,000
Testing	378,300	378,300	378,300
250 V DC Switchboard	27,500	27,500	27,500
250 DC Panelboard	3,600	3,600	3,600
Station Battery & Rack	53,000	53,000	53,000
Battery Chargers	56,500	56,500	56,500
Cable Tray	82,000	82,000	82,000
600 V Instrument Cable	125,100	117,660	117,660
600 V Control Cable	251,600	236,900	236,900
Grounding Systems	370,500	370,500	370,500
480 V Valve Control Center	25,600	25,600	25,600
Conduit-Fittings	158,900	150,000	150,000
600 V Power Cable	80,755	76,355	76,355
1000 V Power Cable	39,580	39,580	39,580
16000 A Isolated Phase Bus	426,100	426,100	426,100
2000 A Segregated Phase Bus	348,000	348,000	348,000
5 KV Switchgear	156,900	161,300	161,300
5 KV Power Cable	33,279	35,379	35,379
Total	$5,923,024	$5,626,254	$5,626,254

TABLE 86

BREAKDOWN OF FPC ACCOUNT 346 - MISCELLANEOUS POWERPLANT EQUIPMENT

FPC Account Number 346 Miscellaneous Powerplant Equipment	First Generation	Second Generation	Third Generation
Laboratory and Sampling Equipment	$ 20,000	$ 20,000	$ 20,000
Tools, Shop, Stores & Work Equipment	125,000	125,000	125,000
Lockers	3,000	3,000	3,000
Emergency Equipment	10,000	10,000	10,000
Miscellaneous Cranes & Hoists	30,000	30,000	30,000
Portable Fire Extinguishers	20,000	20,000	20,000
Communication Equipment	50,000	50,000	50,000
Lunch Room Equipment	20,000	20,000	20,000
Office Furniture & Machines	15,000	15,000	15,000
Total Account 346	$293,000	$293,000	$293,000

TABLE 87

BREAKDOWN OF FPC ACCOUNT 353 - MISCELLANEOUS STATION EQUIPMENT

FPC Account Number 353 Miscellaneous Station Equipment	First Generation	Second Generation	Third Generation
345 KV Main Oil C/B	$ 201,000	$ --	$ --
132 KV Outdoor Switchgear	179,300	--	--
705 MVA Auto-Transformers	829,000	--	--
370 MVA Transformer	--	--	557,000
410 MVA Transformer	--	--	1,000,000
400 MVA Transformer	--	990,000	--
450 MVA Transformer	--	619,000	--
228 MVA Transformer	1,059,000	--	--
506 MVA Transformer	669,000	--	--
Total Account 353	$2,937,000	$1,609,000	$1,557,000

TABLE 88

CAPITAL COST SUMMARY FOR ADVANCED COGAS POWER SYSTEMS

Federal Power Commission Acct. No.		First Generation	Second Generation	Third Generation
340	Land & Land Rights	$ 225,000	$ 225,000	$ 225,000
341	Structures & Improvements	8,492,400	7,742,500	7,742,500
343	Prime Movers (Gas Turbines)	18,871,000	15,329,000	15,978,000
344	Electric Generators (for Gas Turbines)	5,640,000	5,960,000	6,620,000
312	Boiler Plant Equipment	19,560,000	14,199,700	11,115,200
314	Steam Turbine-Generator Units	15,755,500	14,568,200	12,460,500
345	Accessory Electrical Equipment	5,923,000	5,626,300	5,626,300
346	Miscellaneous Powerplant Equipment	293,000	293,000	293,000
353	Misceallenous Station Equipment	2,937,300	1,609,000	1,557,000
	Sub-Total	$ 77,697,200	$65,552,700	$61,587,500
	Other Expenses	1,250,000	1,250,000	1,250,000
	Total Direct Cost	78,947,200	66,802,700	62,837,500
	Engineering, Design, Construction Supervision, and Contingency	11,198,000	10,590,300	10,392,500
	Sub-Total	90,145,000	77,393,000	73,230,000
	Escalation	8,466,000	7,449,000	7,338,000
	Sub-Total	98,611,000	84,842,000	80,568,000
	Interest During Construction	10,650,000	9,163,000	8,701,000
	Total Estimated Cost	$109,261,000	$94,005,000	$89,269,000

TABLE 89

SUMMARY OF POTASSIUM TOPPING CYCLE PERFORMANCE

Turbine Efficiency - 75%
Pump Efficiency - 70%

Cycle	Condition Specified	Potassium Condenser Temperature, T_1 - F	Topping Cycle Thermal Efficiency, η_K - %					
			Potassium Inlet Temperature, T_3 - F					
			1400	1500	1600	1800	2000	2200
Simple Rankine	$X_4 = 0.88$	(as shown)	13.8 (1040 F)(1)	13.7 (1120 F)	13.7 (1200 F)	13.9 (1350 F)	14.6 (1480 F)	15.0 (1600 F)
	T_1	1040	13.8	*(2)	*	*	*	*
		1080	12.2	*	*	*	*	*
		1120	10.5	13.7	*	*	*	*
Reheat	T_1 and $P_4' = \sqrt{P_1P_3}$	1040	13.4	16.3	18.5	22.1	*	*
		1080	11.9	14.8	17.0	20.8	*	*
		1120	10.1	13.1	15.5	19.7	23.0	*
	T_1 and $X_4' = 0.12$	1040	--(3)	--	--	22.1	25.7	28.6
		1080	--	--	--	20.8	24.4	27.3
		1120	--	--	--	19.4	23.0	26.0
	T_1 and $X_7 = 0.12$	1040	--	--	18.8	23.6	26.8	29.2
		1080	--	--	--	22.5	25.7	28.1
		1120	--	--	--	21.3	24.6	27.0
Regenerative (moisture separation)	T_1 and $X_4' = 0.12$	1040	--	--	--	23.5	27.3	30.1
		1080	--	--	--	22.3	26.2	29.1
		1120	--	--	--	20.8	24.9	27.9
	T_1 $X_5 = 0.12$	1040	--	--	--	23.1	26.8	29.5
		1080	--	--	--	21.8	25.6	28.4
		1120	--	--	--	20.3	24.2	27.0

(1) Numbers in parentheses indicate values of T_1.
(2) * Potassium moisture would exceed 12%.
(3) -- Incompatible with the specified cycle configuration.

TABLE 90

BOTTOMING CYCLE WORKING FLUID PROPERTIES

Working Fluid	Steam	Ammonia	Thiophene	R11	R12	R114
Formula	H_2O	NH_3	C_4H_4S	CCl_2F	CCl_2F_2	$CClF_2-CClF_2$
Molecular Weight	18	17.03	84	137.38	120.93	170.93
Boiling Point at 1 atm, F	212	-28.0	182	74.78	-21.6	38.39
Latent Heat of Vaporization at 1 atm, Btu/lb	970.3	589.3	162	78.3	71.0	59.0
Critical Temperature, F	705	271.4	584	388.4	233.6	294.3
Critical Pressure, psia	3205	1657	826	635	597	474
Cost, $/lb	0	0.05	0.25 to 0.50	0.21	0.20	0.65

TABLE 91

COMPARISON OF WORKING FLUIDS FOR A REGENERATIVE CLOSED-CYCLE GAS TURBINE

Gas	Air	Nitrogen	Neon	Argon	80% A 20% He	60% A 40% He	45% A 55% He	40% A 60% He	20% A 80% He	Helium
Properties										
Molecular Weight	28.92	28.02	20.18	39.94	32.75	25.56	20.18	18.37	11.19	4.003
Specific Heat, Btu/lb-F@ 70 F	0.240	0.248	0.250	0.124	0.151	0.194	0.248	0.270	0.444	1.242
Density, lb/cu ft @ 70 F, 1 atm	0.075	0.072	0.052	0.103	0.846	0.065	0.052	0.047	0.028	0.010
Ratio of Specific Heats @ 70 F, 1 atm	1.40	1.40	1.64	1.670	1.669	1.669	1.669	1.669	1.668	1.664
Viscosity, poises x 10^7 @ 70 F, 1 atm	1830	1770	3120	2227	2274	2318	2340	2346	2303	1963
Thermal Conductivity, Btu/hr-ft-F, @ 70 F, 1 atm	15.10	14.00	26.90	10.05	18.72	27.97	35	38.47	52.48	85.90
System Flow Rate (ratio), w	1.0	0.97	0.95	1.93	1.58	1.24	0.97	0.88	0.54	0.19
System Components(1)										
Main Heat Exchanger(2)										
Capacity Rate (ratio), (wc_p)	1.0	1.0	0.99	1.01	1.0	1.0	1.0	1.0	1.0	0.99
Gas Side Flow Length (ratio)	1.0	1.01	0.93	1.06	0.74	0.69	0.68	0.69	0.73	0.92
Number of Tubes (ratio)	1.0	0.99	1.09	1.66	0.94	0.75	0.66	0.64	0.55	0.45
Heat Transfer Area (ratio)	1.0	0.99	1.09	1.66	0.94	0.75	0.66	0.64	0.55	0.45
Gas Side Frontal Area (ratio)	1.0	0.98	1.17	1.57	1.27	1.09	0.96	0.92	0.75	0.49
Turbomachinery										
Optimum Pressure Ratio for Maximum Efficiency	3.0	3.0	2.35	2.25	2.25	2.26	2.26	2.27	2.28	2.30
No. of Stages (ratio)(3)	1.0	1.03	1.04	0.52	0.63	0.81	1.03	1.13	1.85	5.19
Compressor Flow Area (ratio)	1.0	0.99	1.36	1.40	1.39	1.41	1.41	1.39	1.41	1.37
Cycle Thermal Efficiency, %	45.0	45.0	44.0	43.9	43.9	43.9	43.9	43.9	43.9	43.9

(1) All systems designed for equal power, maximum pressure, and maximum temperatures

(2) Exchanger designed for equal gas side pressure drop and negligible heat flow resistance for wall and liquid metal

(3) Constant circumferential velocity in turbomachinery

TABLE 92

COMPOSITION OF PROSPECTIVE HEAT EXCHANGER ALLOYS

Common Name	ASME Spec. Number	Fe	Cr	Ni	Co	Mo	Mn	W	C	Si	Others
Pipe Steel	SA53A	BAL	-	-	-	-	-	-	.13	-	
Pipe Steel	SA53B	BAL	-	-	-	-	-	-	.13	-	
Carbon Steel	SA106A	BAL	-	-	-	-	.60	-	.25	.10	
Carbon Steel	SA106B	BAL	-	-	-	-	.67	-	.30	.10	
Manganese Moly Steel	SA302B	BAL	-	-	-	.5	1.3	-	.20	.20	
Alloy Steel	5A199T11	BAL	1.5	-	-	.5	.45	-	.15 max	1.00 max	
Crolloy P5	SA335P5	BAL	5.0	-	-	.5	.45	-	.15 max	.50 max	
Crolloy P22	SA335P22	BAL	2.25	-	-	1.0	.45	-	.15 max	.50 max	
Stainless Steel 304	SA312TP304	BAL	19.0	9.5	-	-	2.00	-	0.8 max	.75 max	
Stainless Steel 316	SA312TP316	BAL	17.0	12.5	-	2.5	2.00	-	0.8 max	.75 max	
Tool Steel H11	None	BAL	5.00	-	-	1.5	-	-	.35	-	.4 V
Haynes 56	None	BAL	21.0	13.0	11.5	4.5	1.50	1.5	.27	1.00 max	.75 (Cb + Ta)
Haynes 25	None	3.00	20.0	10.0	BAL	-	1.50	15.0	.10	1.00 max	
Waspalloy	None	1.00	19.0	BAL	14.0	4.0	-	-	.10	-	tr B, Zr, 3.0 Ti, 1.3 Al
Udimet 700	None	4.00	15.0	BAL	19.0	5.0	-	-	.10	-	.03 B, 3.5 Ti, 4.5 Al
TD Nickel	None	-	-	BAL	-	-	-	-	-	-	2.0 Tho3
Hastelloy X	None	18.00	21.00	BAL	2.00	9.00	-	1.0	.10	-	
Nimonic 75	None	0.5	20.5	BAL	-	-	-	-	.10	-	.15Al,.35Ti
Columbium 1Zr	None	-	-	-	-	-	-	-	.02	-	1.0 Zr, Cb BAL
Columbium F48	None	-	-	-	-	5.0	-	15.0	.05	-	1.0 Zr, Cb BAL
Advanced Columbium	None			Not	Available						

TABLE 93

SPECIFICATIONS OF TURBOMACHINERY FOR HELIUM POWER SYSTEMS

	High-Temperature System	Low-Temperature Syste
Operating Conditions		
Turbine Inlet Temp, F	1600	1200
Condenser Temp., F	90	90
Turbine Inlet Pressure, psia	990	785
Turbine Outlet Pressure , psia	310	310
Thermodynamic Cycle Efficiency, %	48.5	42.3
Net Power System Efficiency , %	41.4	36.1
Unit Output , Mw	500	250
Component Specifications		
Compressor		
RPM	3600	5500
No. Stages (Total for 3 Cases)	22	17
Mean Diameter (max), in.	54	38
Isentropic Efficiency, %	89	89
High Turbine(1)		
RPM	3600	5500
No. Stages	26(2)	5
Mean Diameter (max) , in.	65.5	40
Isentropic Efficiency , %	91	91
Low Turbine(1)		
RPM	--	3600
No. Stages	--	16(2)
Mean Diameter (max) in.	--	67
Isentropic Efficiency %	--	71

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(1) For the high-temperature cycle, the compressor drive (high) turbine and the power (low) turbine are not in separate cases.

(2) For dual-exhaust ends.

TABLE 94

SPECIFICATIONS OF TURBOMACHINERY FOR CARBON DIOXIDE POWER SYSTEMS

	High Temperature System		Low Temperature System	
Operating Conditions				
Turbine Inlet Temp., F	1600		1200	
Condenser Temp., F	100		100	
Turbine Inlet Pressure, psia	2940		2940	
Turbine Outlet Pressure, psia	294		294	
Thermodynamic Cycle Eff., %	44.7		44.6	
Net Power System Efficiency, %	38.2		38.1	
Unit Output, Mw	500		500	
Component Specifications				
Compressor	High(1)	Low	High(1)	Low
RPM	7200	3600	7200	3600
No. Stages	6	19	6	19
Mean Diameter (max), in.	27.8	36.8	27.8	36.8
Isentropic Efficiency, %	85	85	85	85
High Turbine				
RPM	3600		3600	
No. Stages	7		2	
Mean Diameter (max), in.	43.8		34.6	
Isentropic Efficiency, %	90		90	
Low Turbine				
RPM	--		3600	
No. Stages	--		4	
Mean Diameter (max) in.	--		38.9	
Blade Height (inlet/outlet)	--		4.20/8.88	
Isentropic Efficiency	--		90	

---

(1) This cycle has a split stream in which all the fluid is compressed in the low compressor, a fraction diverted and cooled to the subcooled liquid state, then pumped to the highest cycle pressure where it is vaporized and mixed with the remaining fluid fraction which has been compressed in the high compressor.

TABLE 95

SPECIFICATIONS OF TURBOMACHINERY FOR SULFUR DIOXIDE POWER SYSTEMS

	Low Temperature System	High Temperature System
Operating Conditions		
Turbine Inlet Temp , F	1200	1600
Condenser Temp, F	100	100
Turbine Inlet Pressure , psia	3500	3500
Turbine Outlet Pressure, psia	84	84
Thermodynamic Cycle Efficiency, %	42.5	57.7
Net Power System Efficiency, %	36.3	49.2
Unit Output, Mw	500	500
Component Specifications		
Compressor	(1)	
RPM		7200
No. Stages		15
Mean Diameter (max) , in.		27
Isentropic Efficiency, %		85
High Turbine		
RPM	7200	7200
No. Stages	5	5
Mean Diameter (max) , in.	25	28.8
Isentropic Efficiency, %	90	90
Low Turbine		
RPM	3600	3600
No. Stages	4	4
Mean Diameter (max) , in.	58.3	60
Isentropic Efficiency, %	90	90

---

(1) This cycle has complete condensation; thus there is no compressor.

TABLE 96

SPECIFICATIONS OF HEAT EXCHANGERS FOR HELIUM POWER SYSTEMS

	High-Temperature System	Low-Temperature System
Heater Heat Transfer Area		
Furnace, ft^2	108,000	166,000
Convection Section, ft^2	390,000	242,000
Air Preheater, ft^2	1,000,000	1,000,000
Regenerator Heat Transfer Area		
High-Temperature Section, ft^2	240,000	341,000
Low-Temperature Section, ft^2	636,000	960,000

TABLE 95

SPECIFICATIONS OF TURBOMACHINERY FOR SULFUR DIOXIDE POWER SYSTEMS

	Low Temperature System	High Temperature System
Operating Conditions		
Turbine Inlet Temp , F	1200	1600
Condenser Temp, F	100	100
Turbine Inlet Pressure , psia	3500	3500
Turbine Outlet Pressure, psia	84	84
Thermodynamic Cycle Efficiency, %	42.5	57.7
Net Power System Efficiency, %	36.3	49.2
Unit Output, Mw	500	500
Component Specifications		
Compressor	(1)	
RPM		7200
No. Stages		15
Mean Diameter (max) , in.		27
Isentropic Efficiency, %		85
High Turbine		
RPM	7200	7200
No. Stages	5	5
Mean Diameter (max) , in.	25	28.8
Isentropic Efficiency, %	90	90
Low Turbine		
RPM	3600	3600
No. Stages	4	4
Mean Diameter (max) , in.	58.3	60
Isentropic Efficiency, %	90	90

---

(1) This cycle has complete condensation; thus there is no compressor.

TABLE 96

SPECIFICATIONS OF HEAT EXCHANGERS FOR HELIUM POWER SYSTEMS

	High-Temperature System	Low-Temperature System
Heater Heat Transfer Area		
Furnace, ft^2	108,000	166,000
Convection Section, ft^2	390,000	242,000
Air Preheater, ft^2	1,000,000	1,000,000
Regenerator Heat Transfer Area		
High-Temperature Section, ft^2	240,000	341,000
Low-Temperature Section, ft^2	636,000	960,000

TABLE 97

SPECIFICATIONS OF HEAT EXCHANGERS FOR CARBON DIOXIDE POWER SYSTEMS

	High-Temperature System	Low-Temperature System
Heater Heat Transfer Area		
Furnace, ft^2	100,000	104,000
Convection Section, ft^2	370,000	178,000
Air Preheater, ft^2	1,000,000	1,000,000
Regenerator Heat Transfer Area, ft^2	1,800,000	3,500,000

TABLE 98

SPECIFICATIONS OF HEAT EXCHANGERS FOR SULFUR DIOXIDE POWER SYSTEMS

	High-Temperature System	Low-Temperature System
Heater Heat Transfer Area		
Furnace, ft^2	70,000	95,000
Convection Section, ft^2	252,000	335,000
Air Preheater, ft^2	1,000,000	1,000,000
Regenerator Heat Transfer Area, ft^2	5,300,000	900,000
Condenser Heat Transfer Area, ft^2	690,000	1,100,000

TABLE 99

CAPITAL COST SUMMARY FOR HELIUM CLOSED-CYCLE GAS TURBINE SYSTEMS

1000-Mw Nominal Output

Federal Power Commission Acct. No.	Description	Low Temperature System	High Temperature System
		Capital Costs - Thousand Dollars	
340	Land and Land Rights	225	225
341	Structures and Improvements	10,100	7,500
342	Heater Plant Equipment	43,200	58,200
343	Prime Mover (includes regenerator)	22,000	18,800
344	Electric Generators	9,700	9,500
345	Accessory Electrical Equipment	4,850	4,500
346	Miscellaneous Powerplant Equipment	215	220
353	Station Equipment	3,800	1,300
	Other Expenses	1,250	1,250
Total Direct Cost		95,340	101,495
	Engineering, Design, Construction Supervision	10,720	1,020
	Escalation During Construction	15,700	16,700
	Interest During Construction	15,250	16,300
	Total Estimated Power System Cost	137,010	145,615

TABLE 100

CAPITAL COST SUMMARY FOR CARBON DIOXIDE POWER SYSTEMS

1000-Mw Nominal Size

Federal Power Commission Acct. No.	Description	Low Temperature System	High Temperature System
		Capital Costs - Thousand Dollars	
340	Land and Land Rights	225	225
341	Structure and Improvements	9,100	9,100
342	Heat Plant Equipment	40,130	63,340
343	Prime Mover (includes regenerator)	43,600	41,000
344	Electrical Generators	9,500	9,500
345	Accessory Electrical Equipment	10,140	10,140
346	Miscellaneous Powerplant Equipment	450	450
353	Station Equipment	1,570	1,570
	Other Expenses	1,250	1,250
Total Direct Costs		115,965	131,605
	Engineering, Design, Construction Supervision	13,800	15,600
	Escalation During Construction	19,300	21,800
	Interest During Construction	18,950	21,100
	Total Estimated Power System Cost	167,965	190,105

TABLE 101

CAPITAL COST SUMMARY FOR SULFUR DIOXIDE POWER SYSTEMS

1000-Mw Nominal Size

Federal Power Commission Acct. No.	Description	Low Temperature System	High Temperature System
		Capital Costs - Thousand Dollars	
340	Land and Land Rights	225	225
341	Structure and Improvements	12,150	12,150
342	Heat Plant Equipment	44,100	69,050
343	Prime Mover (includes regenerator)	30,080	58,360
344	Electrical Generators	9,500	9,500
345	Accessory Electrical Equipment	10,140	10,140
346	Miscellaneous Powerplant Equipment	450	450
353	Station Equipment	1,570	1,570
	Other Expenses	1,250	1,250
Total Direct Costs		109,665	162,255
	Engineering, Design, Construction Supervision	13,050	19,320
	Escalation During Construction	18,320	27,100
	Interest During Construction	17,980	26,700
	Total Estimated Power System Cost	159,015	226,375

FIG. 29

TURBINE INLET TEMPERATURE PROGRESSION FOR BASE-LOAD OPERATION

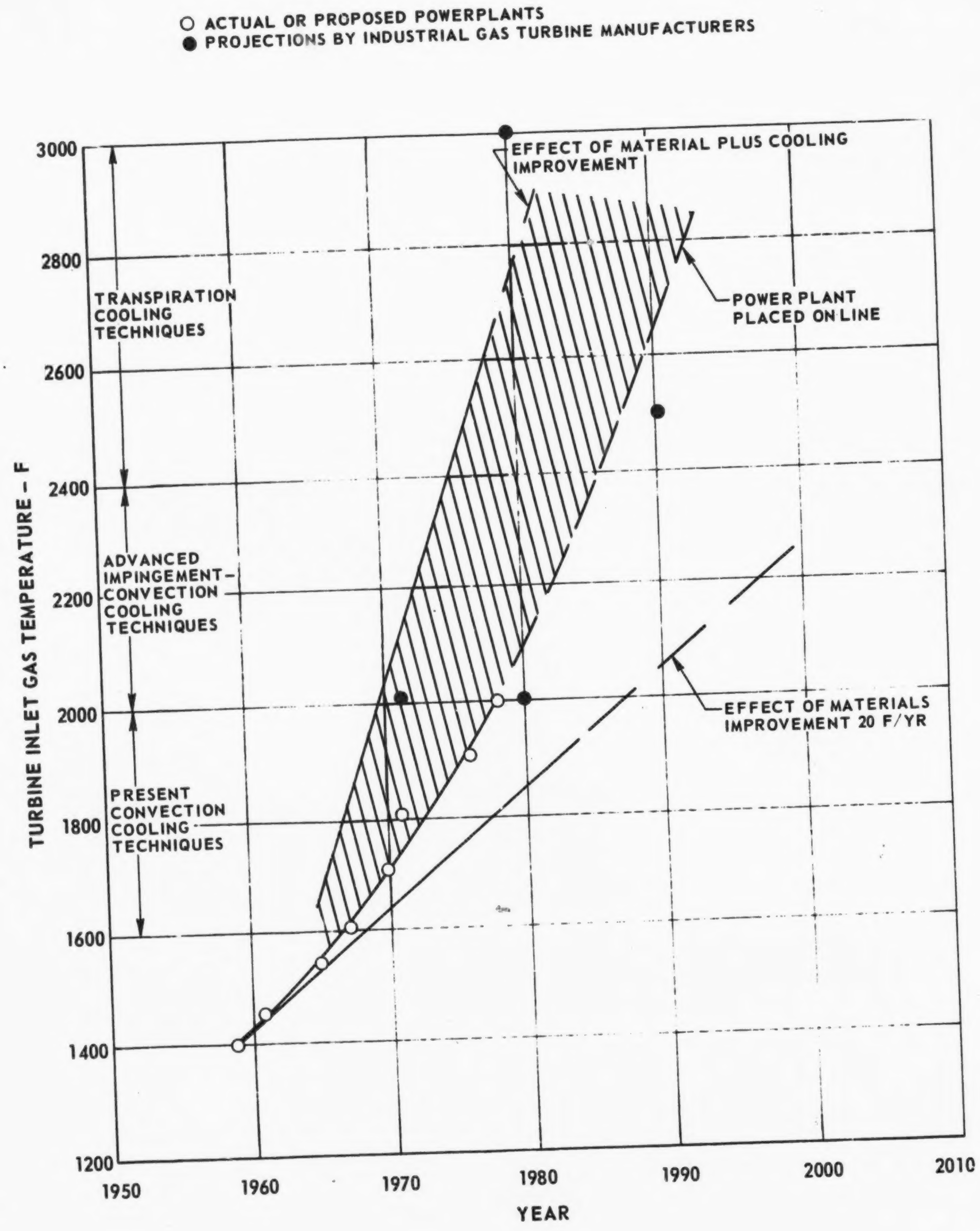


FIG. 30

PROGRESSION OF AIRCRAFT COMPRESSOR TECHNOLOGY

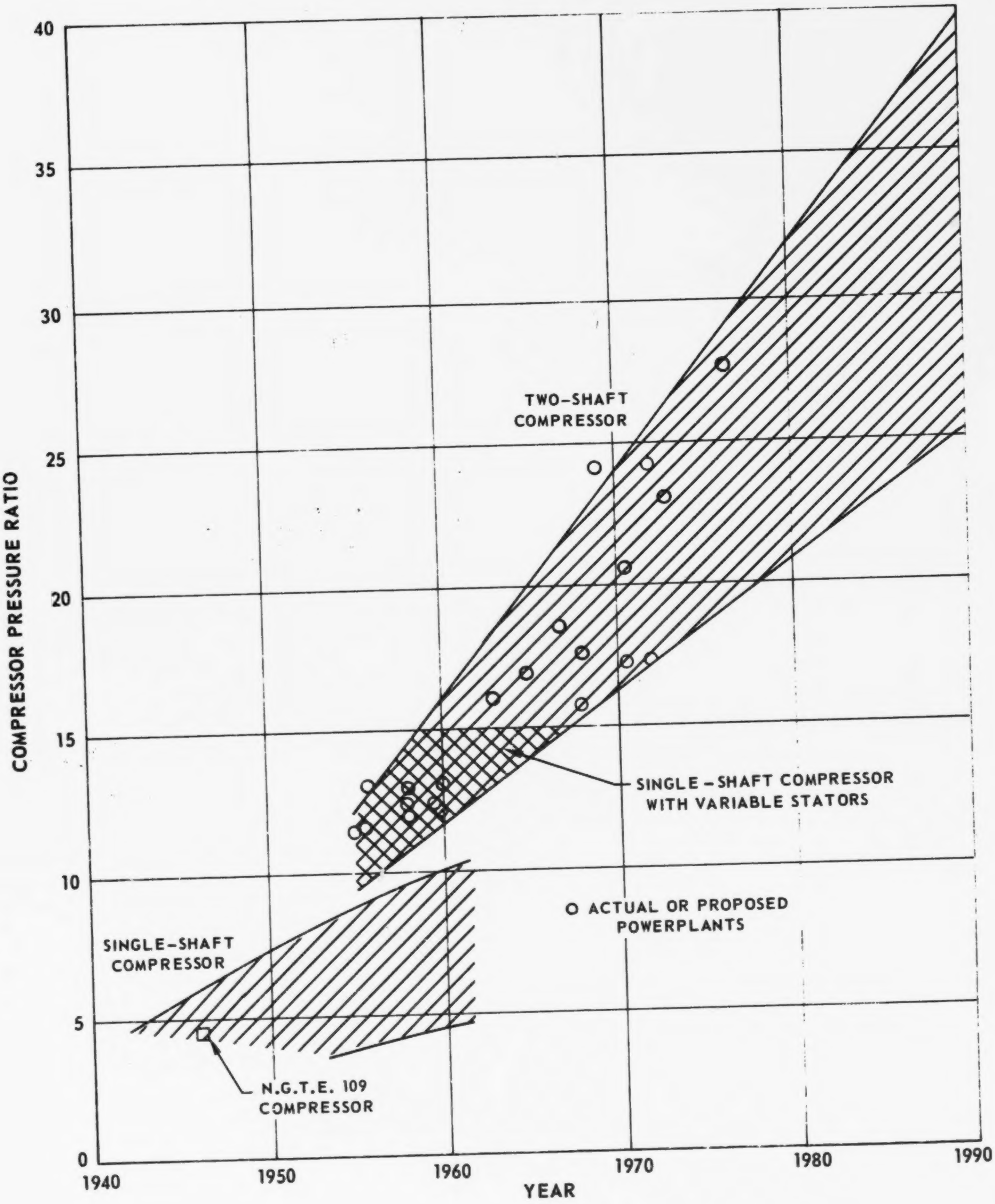
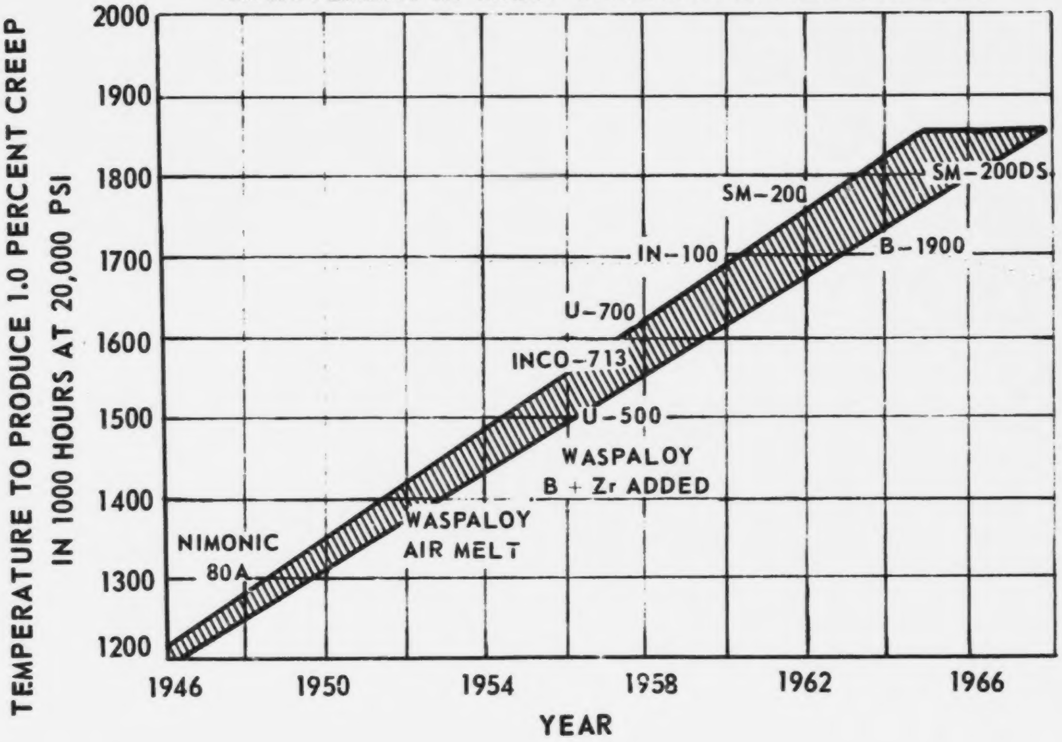


FIG. 31

ADVANCES IN AIRCRAFT TURBINE MATERIALS

IMPROVEMENT IN CREEP STRENGTH IN BLADE ALLOYS



IMPROVEMENT IN CREEP STRENGTH IN VANE ALLOYS

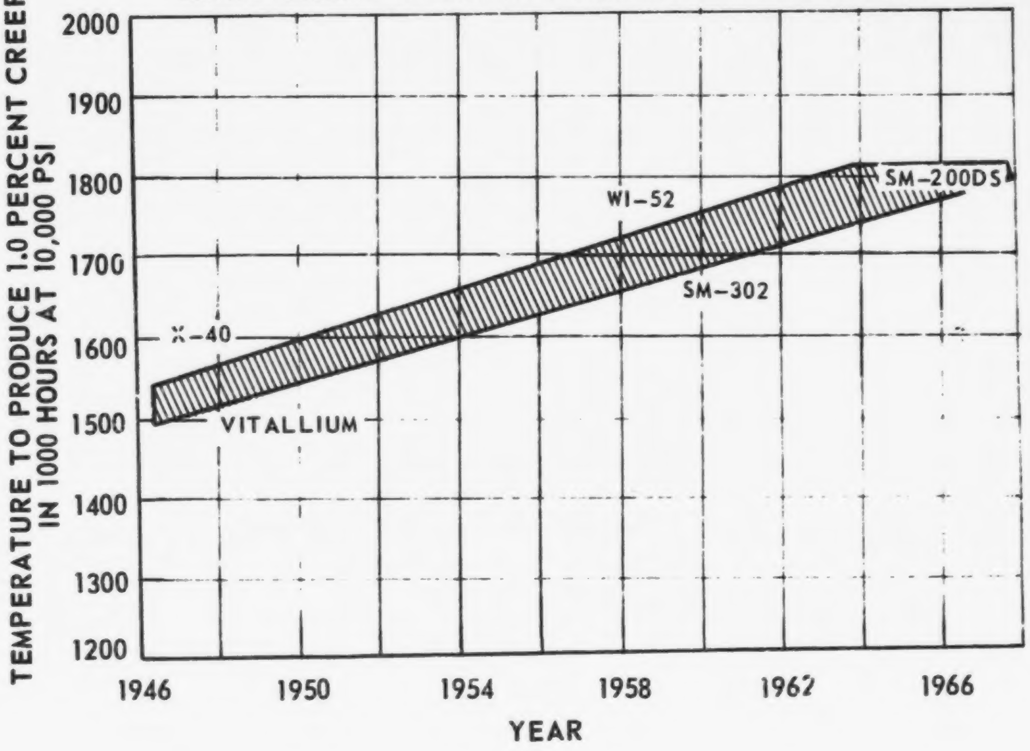


FIG. 32

CREEP STRENGTH FOR ADVANCED TURBINE BLADE MATERIALS

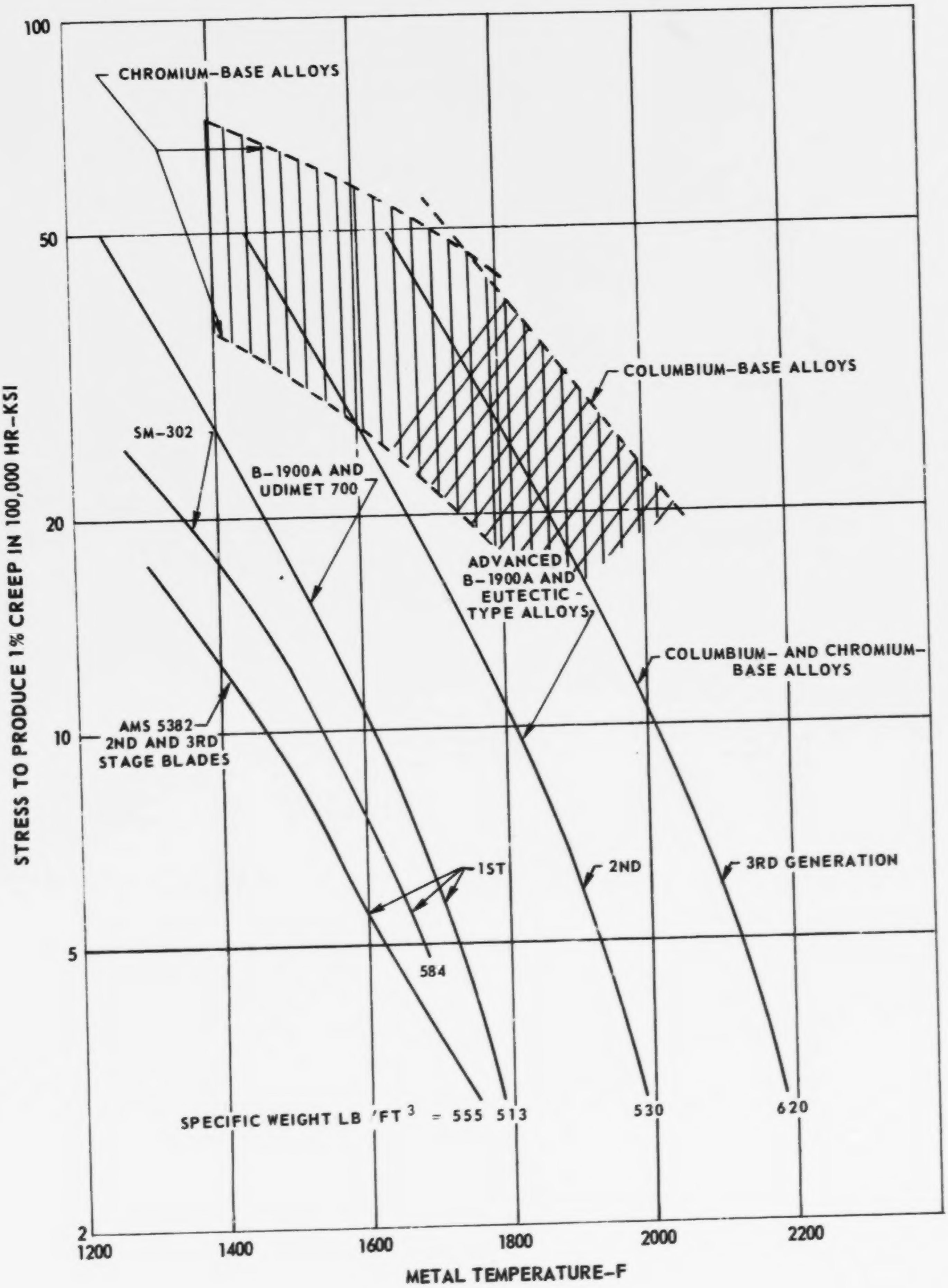
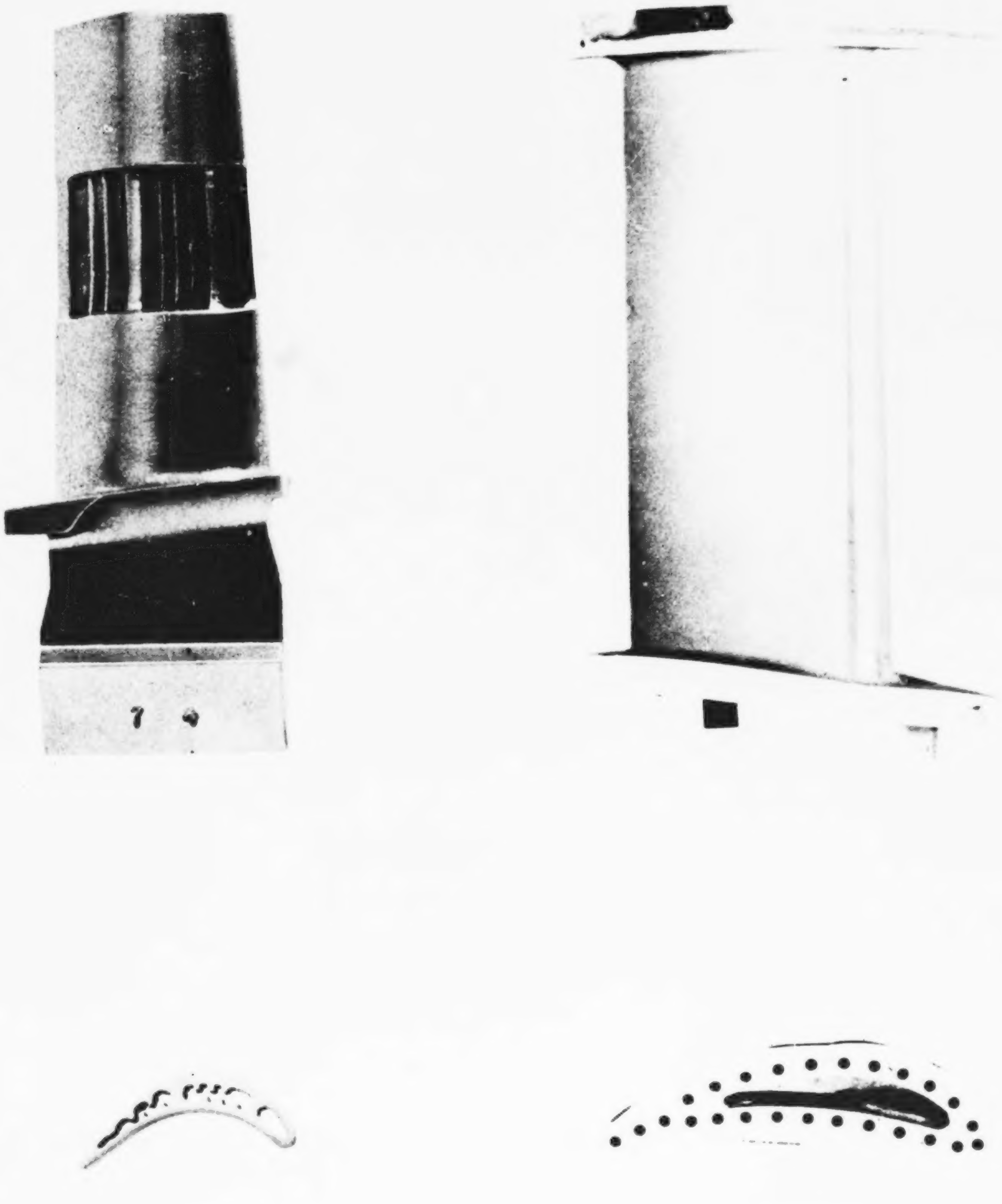


FIG. 33

SIMPLE CONVECTION-COOLED BLADES FOR AIRCRAFT POWERPLANTS

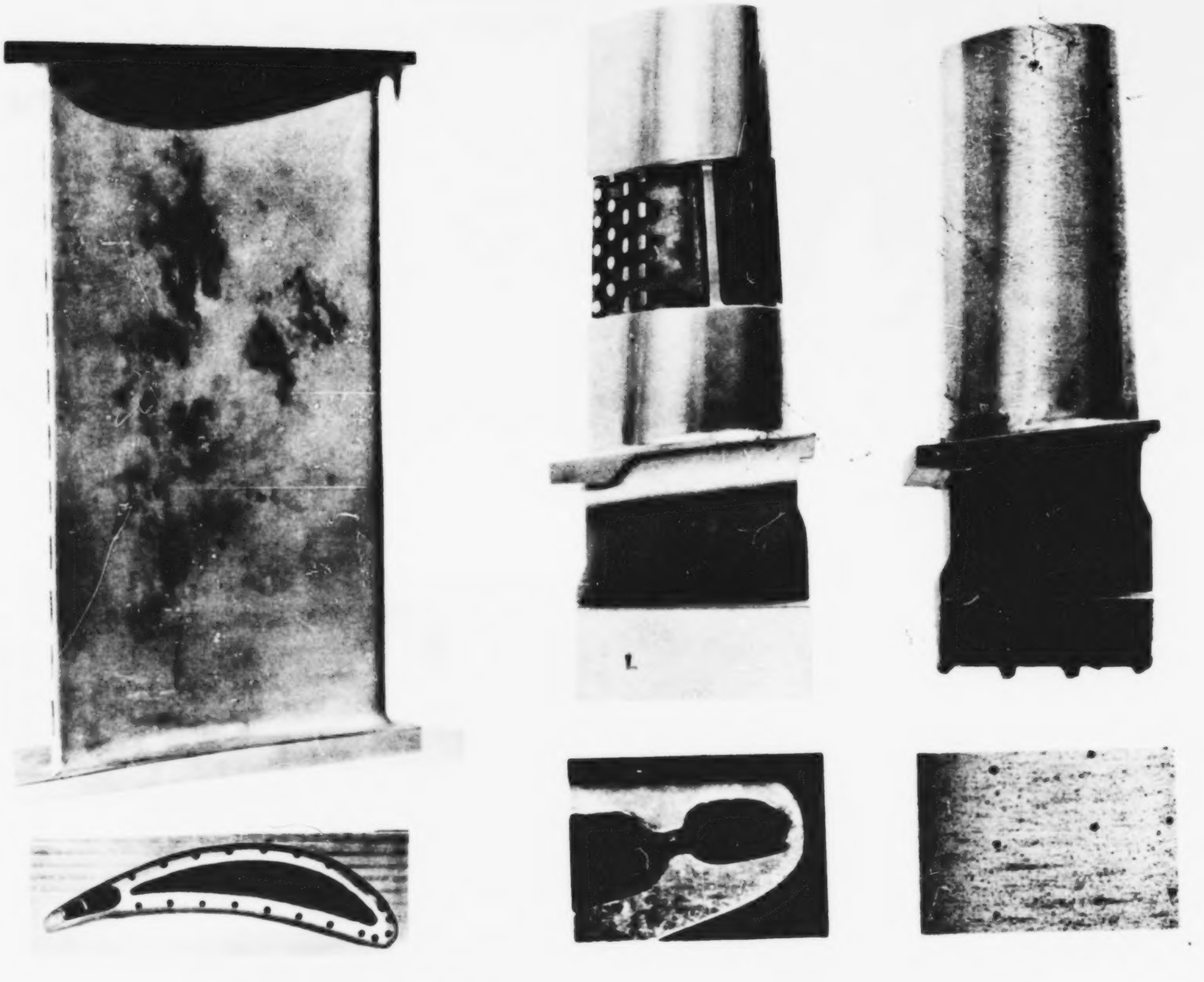


BLADE

VANE

FIG. 34

ADVANCED BLADE COOLING CONFIGURATIONS FOR AIRCRAFT POWERPLANTS

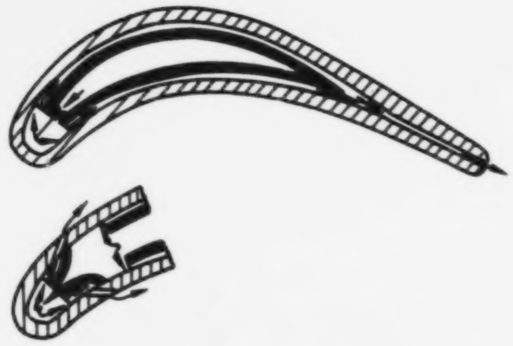


FILM-COOLED

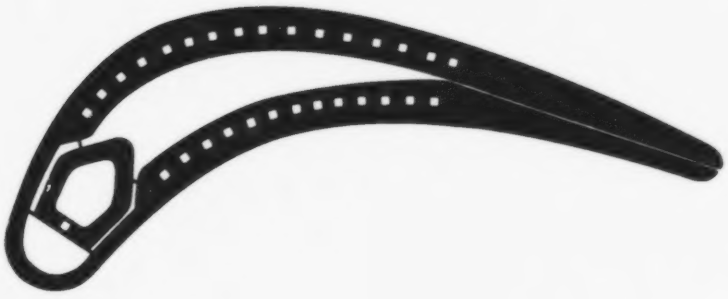
IMPINGEMENT-CONVECTION

TRANSPIRATION COOLED

TYPICAL IMPINGEMENT-CONVECTION COOLED BLADES FOR AIRCRAFT AND STATIONARY POWERPLANTS



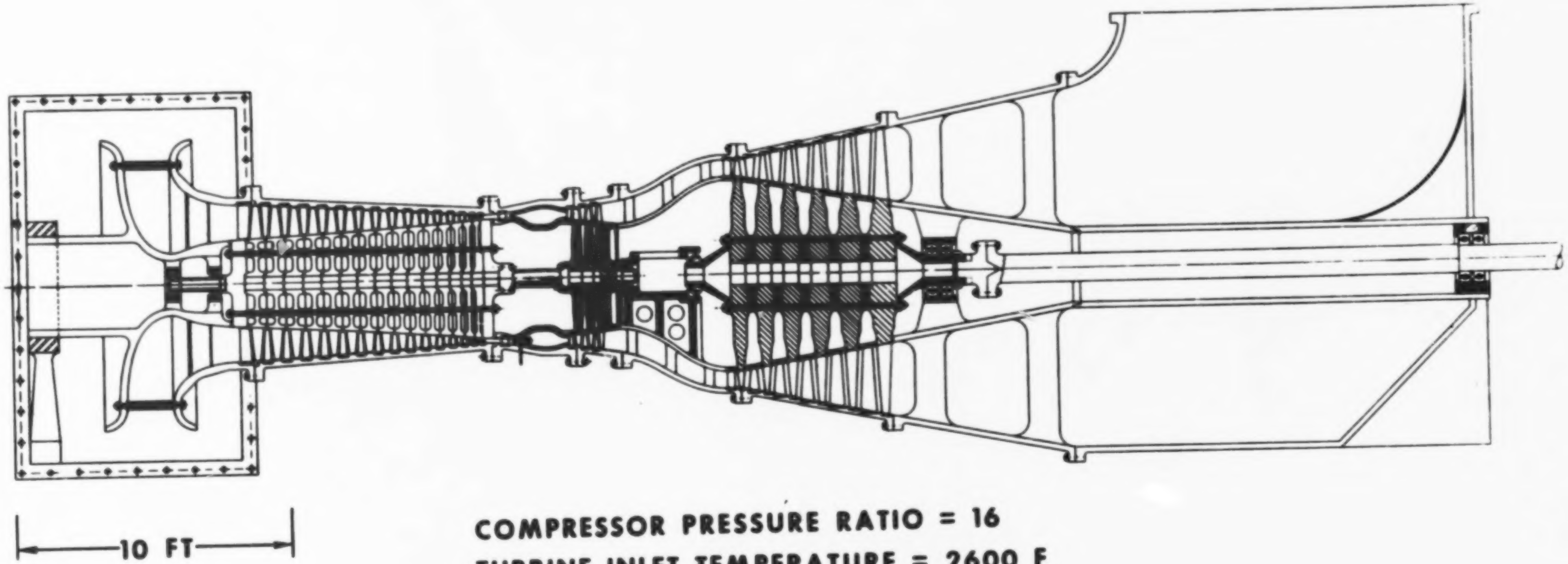
AIRCRAFT POWERPLANT AIR-COOLED BLADE



STATIONARY POWERPLANT AIR-COOLED BLADE

FIG. 36

250-MW BASE LOAD GAS TURBINE



COMPRESSOR PRESSURE RATIO = 16
TURBINE INLET TEMPERATURE = 2600 F
EXHAUST FLOW = 1040 LB/SEC
HEAT RATE = 8760 BTU/KW HR (HHV)

United Aircraft
Research Laboratories

FIG. 37

BASE-LOAD GAS TURBINE FLOW DIAGRAM

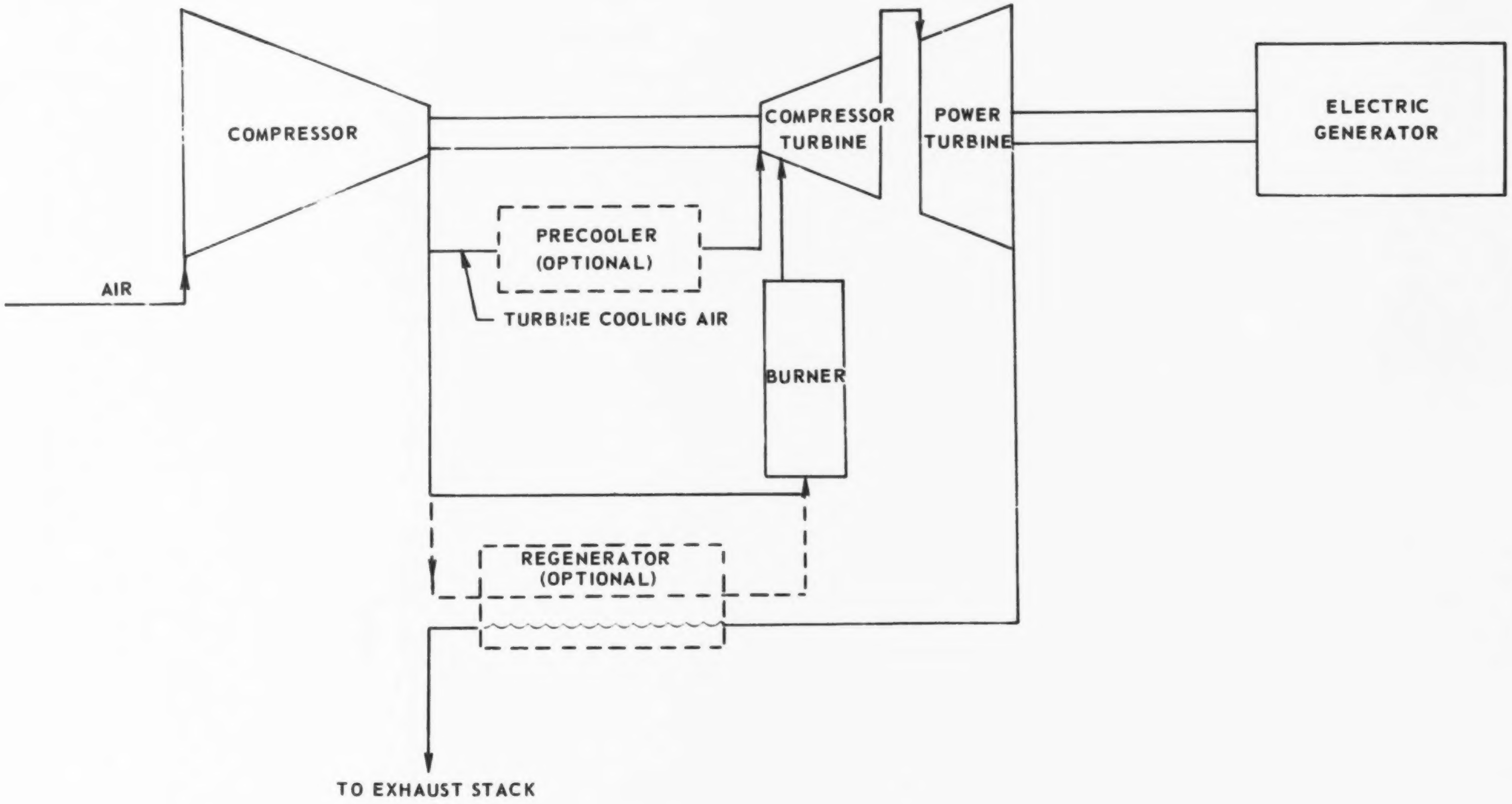


FIG. 38

BASE – LOAD GAS TURBINE COMPUTER PROGRAM FLOW CHART

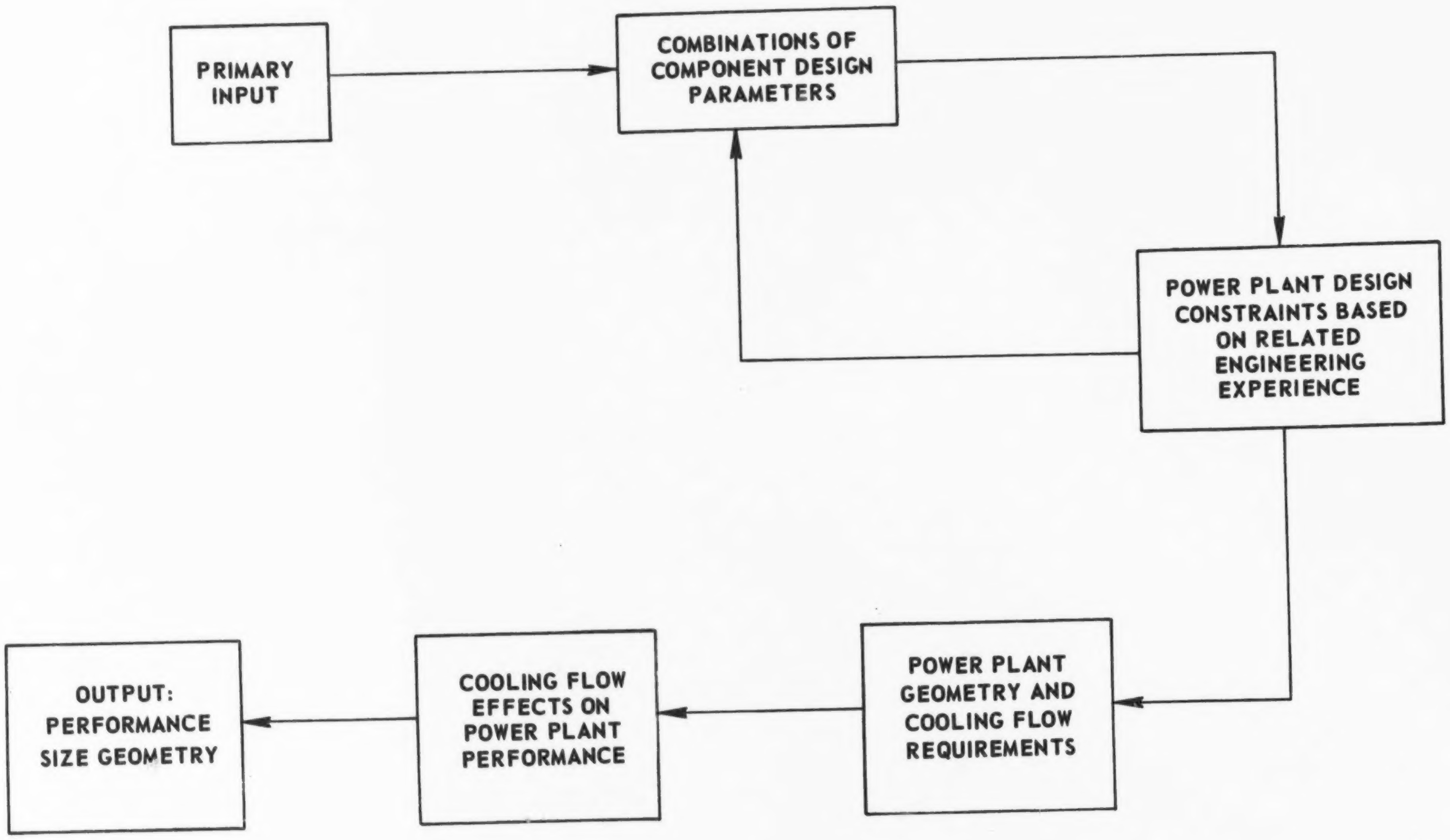

PRIMARY INPUT
COMBINATIONS OF COMPONENT DESIGN PARAMETERS
POWER PLANT DESIGN CONSTRAINTS BASED ON RELATED ENGINEERING EXPERIENCE
POWER PLANT GEOMETRY AND COOLING FLOW REQUIREMENTS
COOLING FLOW EFFECTS ON POWER PLANT PERFORMANCE
OUTPUT: PERFORMANCE SIZE GEOMETRY

FIG. 39

THERMAL EFFICIENCY PERFORMANCE FOR SIMPLE-CYCLE GAS TURBINES

FIRST-GENERATION TECHNOLOGY

FUEL – GAS I (HHV = 162 BTU/FT^3)

AMBIENT – 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT– CONVECTION;
COMPRESSOR BLEED AIR UNCOOLED

ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

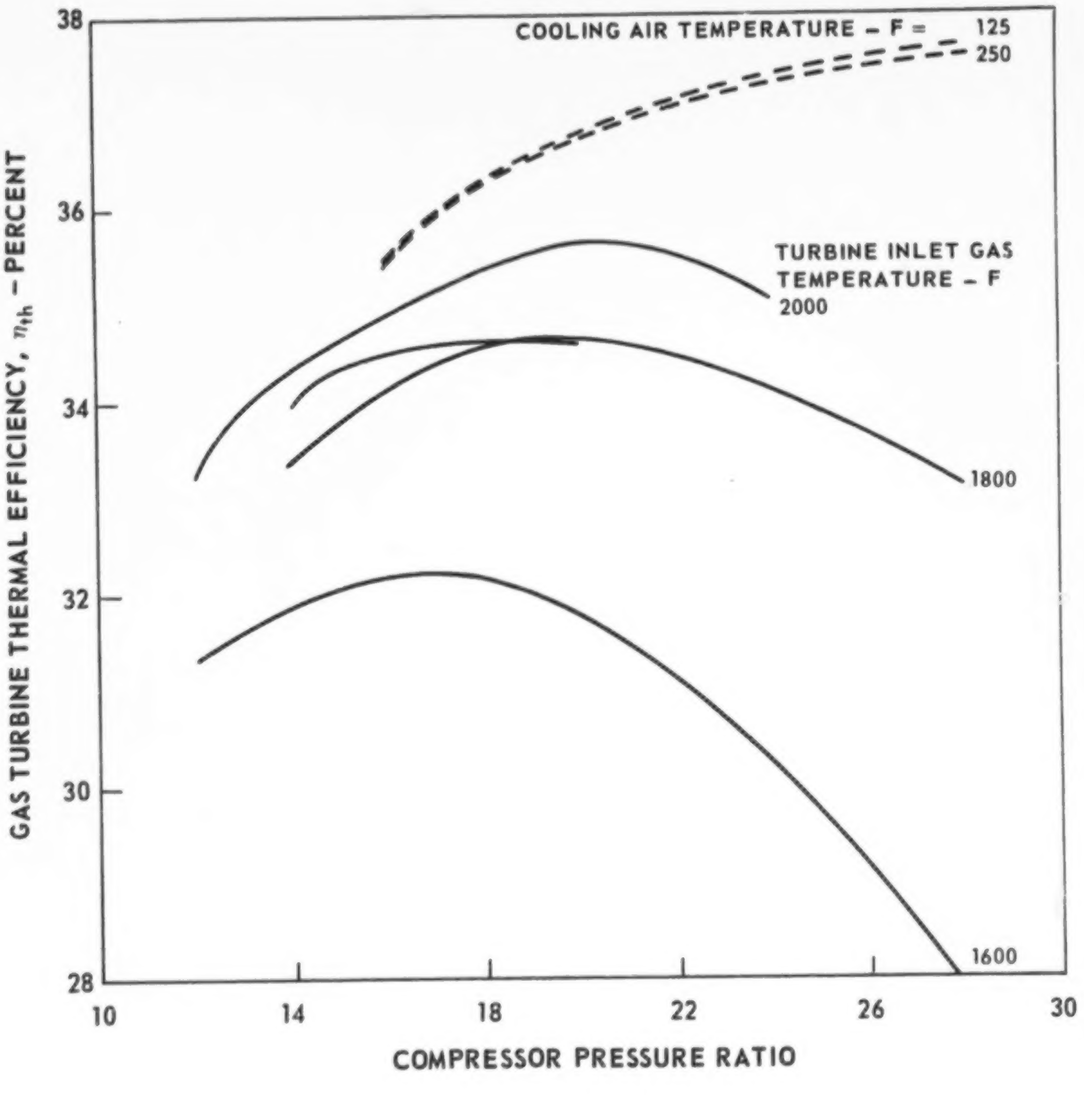


FIG. 40

SPECIFIC OUTPUT PERFORMANCE FOR SIMPLE-CYCLE GAS TURBINES

FIRST-GENERATION TECHNOLOGY
FUEL – GAS I (HHV = 162 BTU/FT^3)

AMBIENT – 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION; COMPRESSOR BLEED AIR UNCOOLED
ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

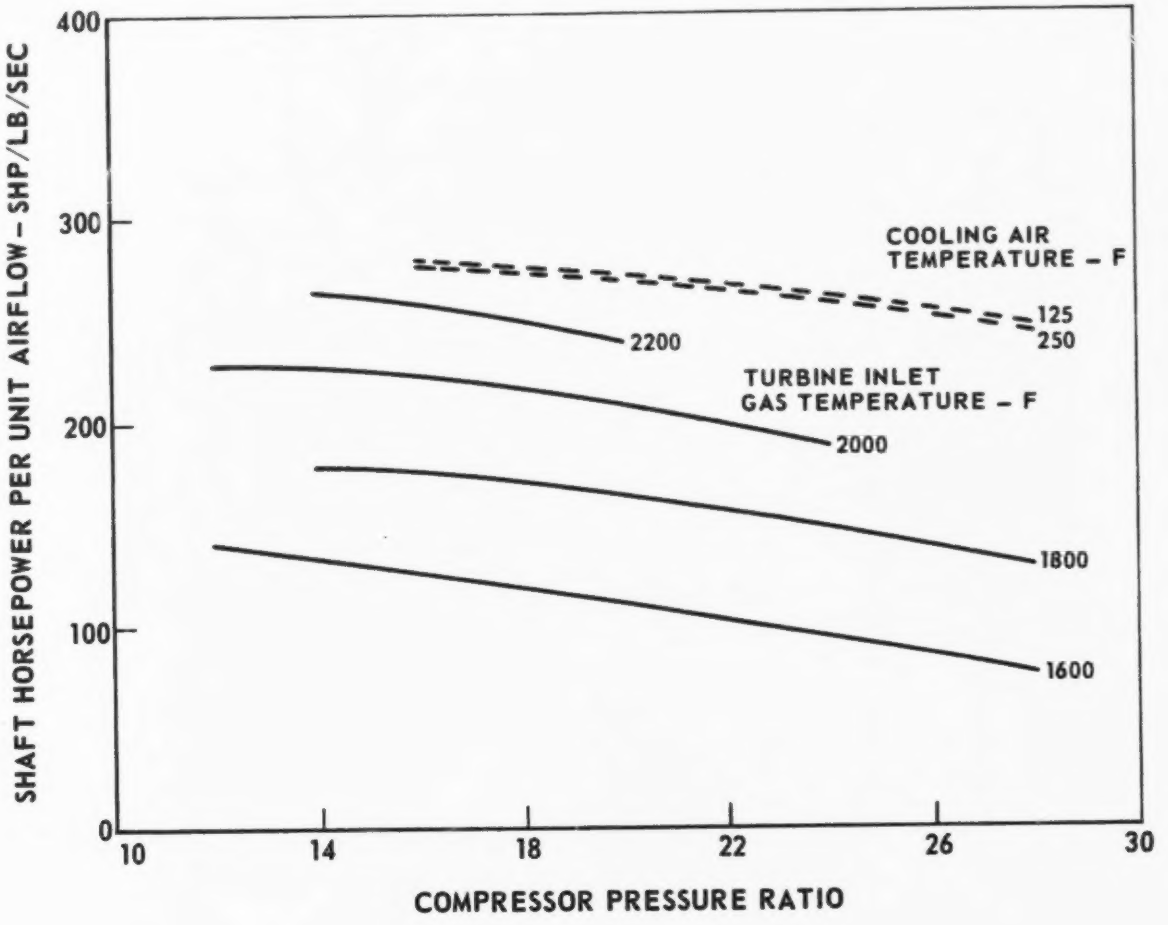


FIG. 41

DEGRADATION IN SIMPLE-CYCLE GAS TURBINE THERMAL EFFICIENCY PERFORMANCE DUE TO TURBINE COOLING

FIRST-GENERATION TECHNOLOGY
FUEL – GAS I (HHV = 162 BTU/FT^3)
AMBIENT – 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION; COMPRESSOR BLEED AIR UNCOOLED

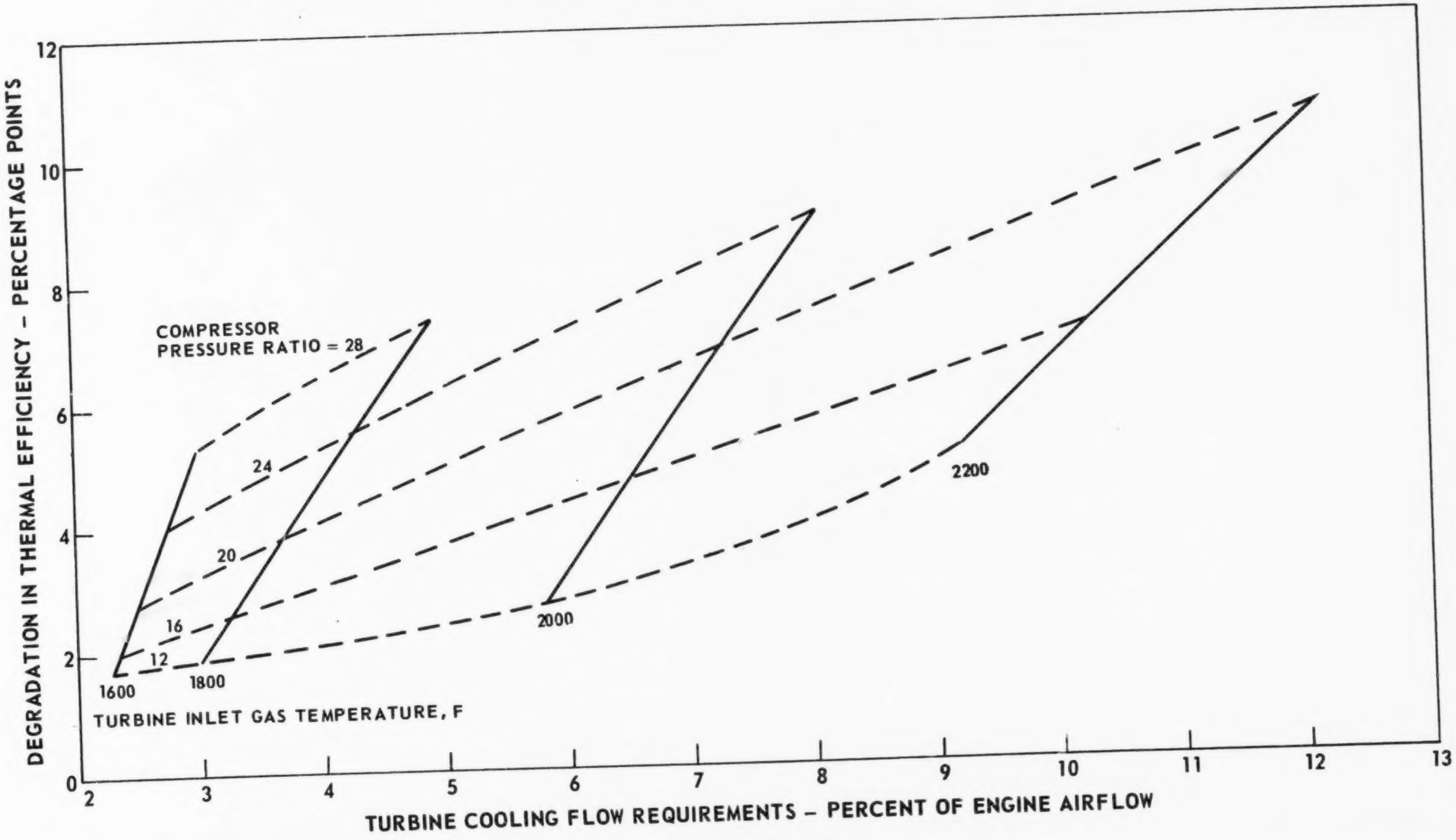


FIG. 42

DEGRADATION IN SIMPLE-CYCLE GAS TURBINE SPECIFIC OUTPUT PERFORMANCE DUE TO TURBINE COOLING

FIRST-GENERATION TECHNOLOGY

FUEL GAS I (HHV = 162 BTU/FT^3)

AMBIENT – 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION; COMPRESSOR BLEED AIR UNCOOLED

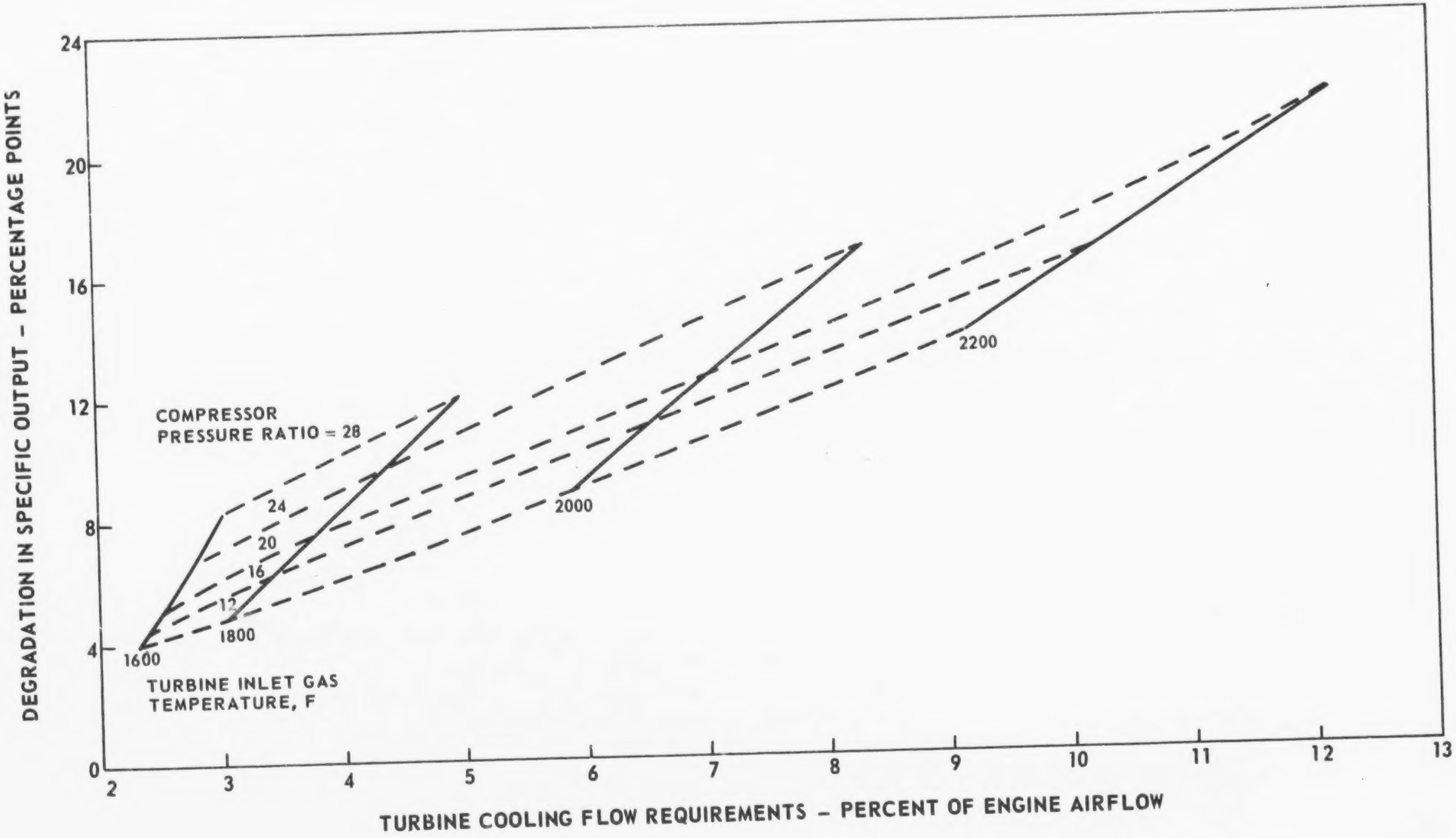


FIG. 43

SIMPLE CYCLE GAS TURBINE PERFORMANCES WITH LOW HEAT - CONTENT FUELS

FIRST-GENERATION TECHNOLOGY
FUEL- GAS I (HHV = 162 BTU/FT^3)
AMBIENT - 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT - CONVECTION
ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

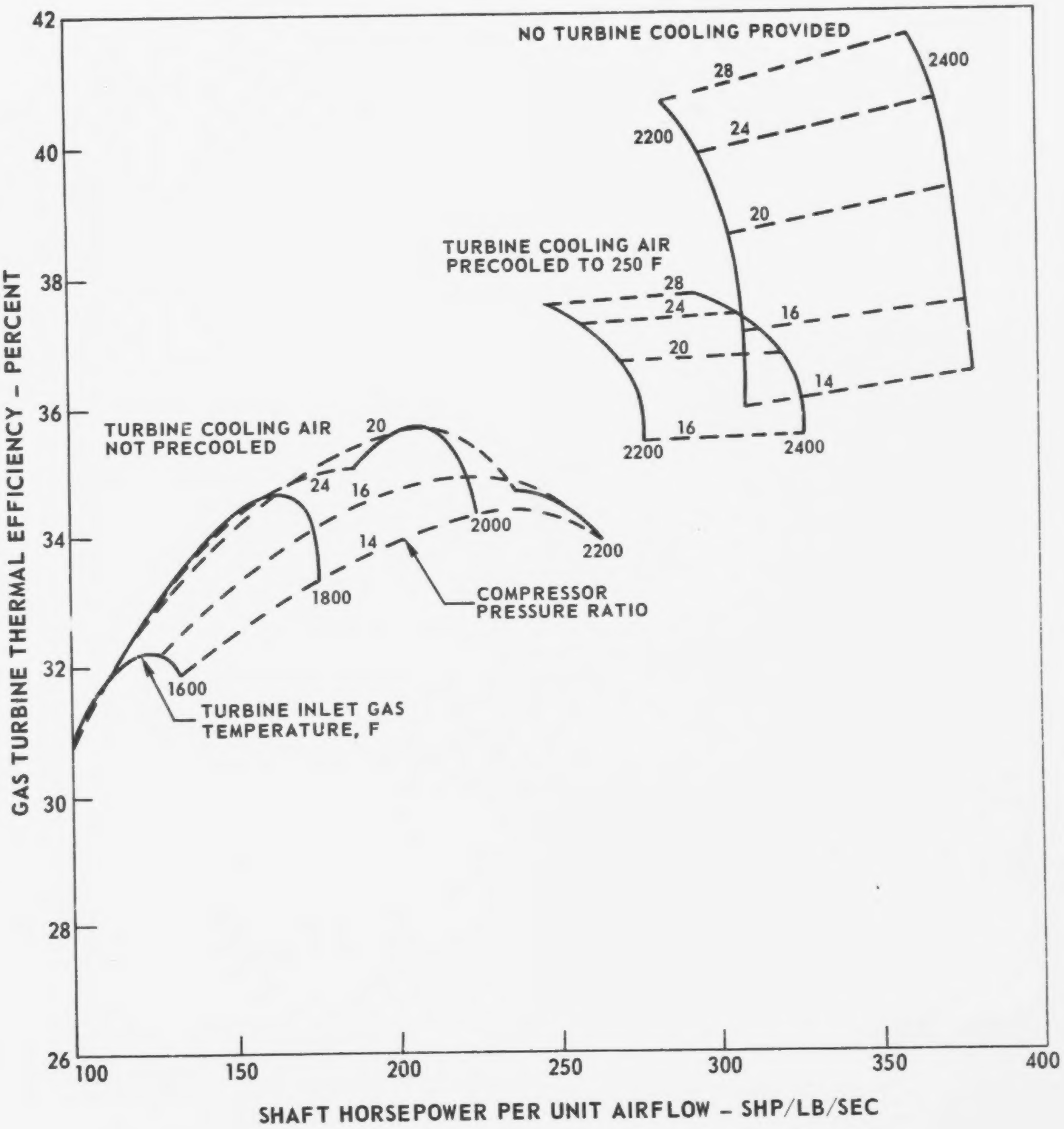


FIG. 44

SIMPLE-CYCLE GAS TURBINE PERFORMANCE WITH LOW HEAT-CONTENT FUELS

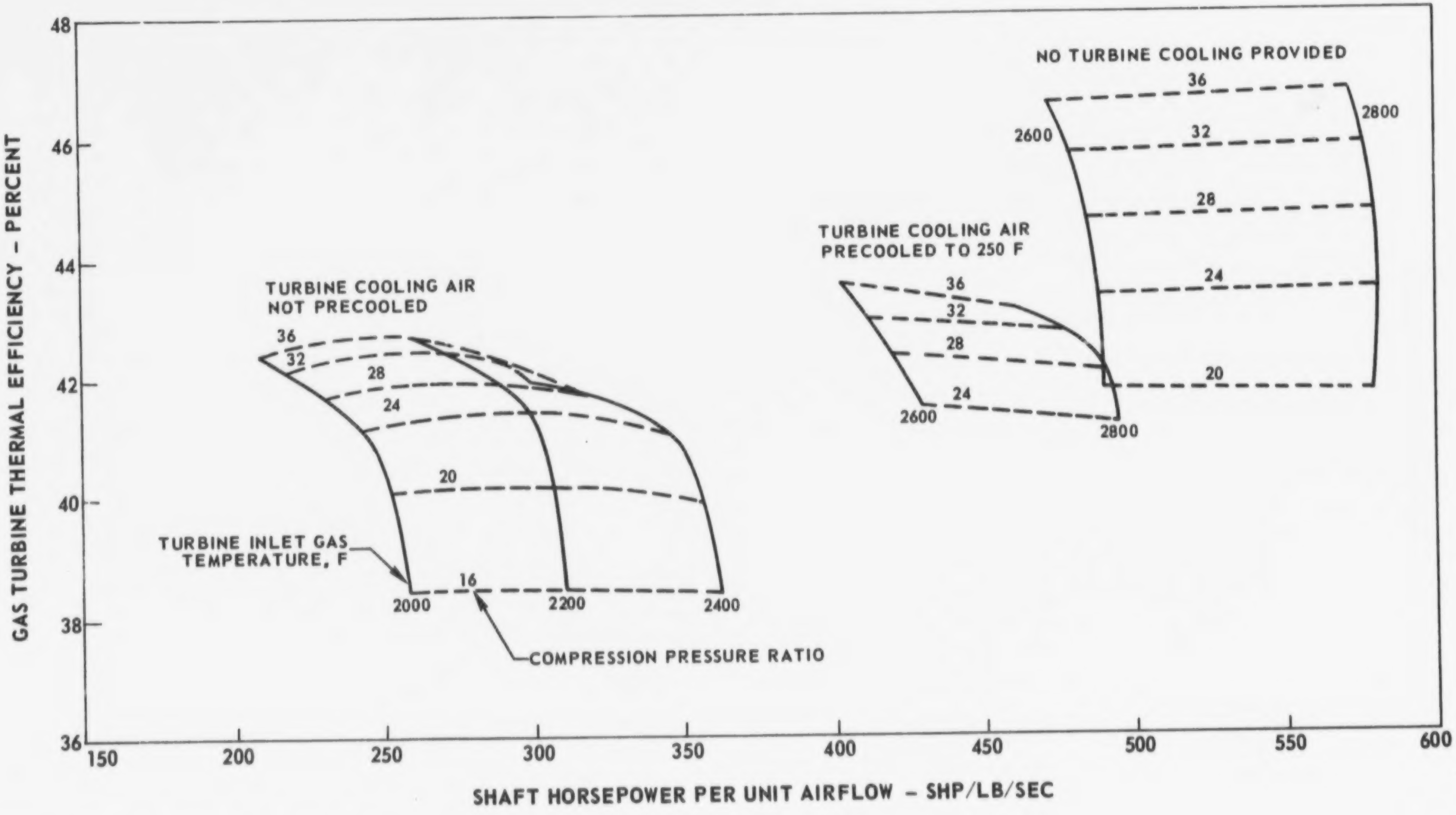
SECOND - GENERATION TECHNOLOGY

FUEL - GAS I (HHV = 162 BTU/FT^3)

AMBIENT - 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION

ELECTRICAL GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED



SIMPLE-CYCLE GAS TURBINE PERFORMANCE WITH LOW HEAT-CONTENT FUELS

THIRD-GENERATION TECHNOLOGY

FUEL – GAS T (HHV = 162 BTU/FT^3)

AMBIENT – 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT – CONVECTION

ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

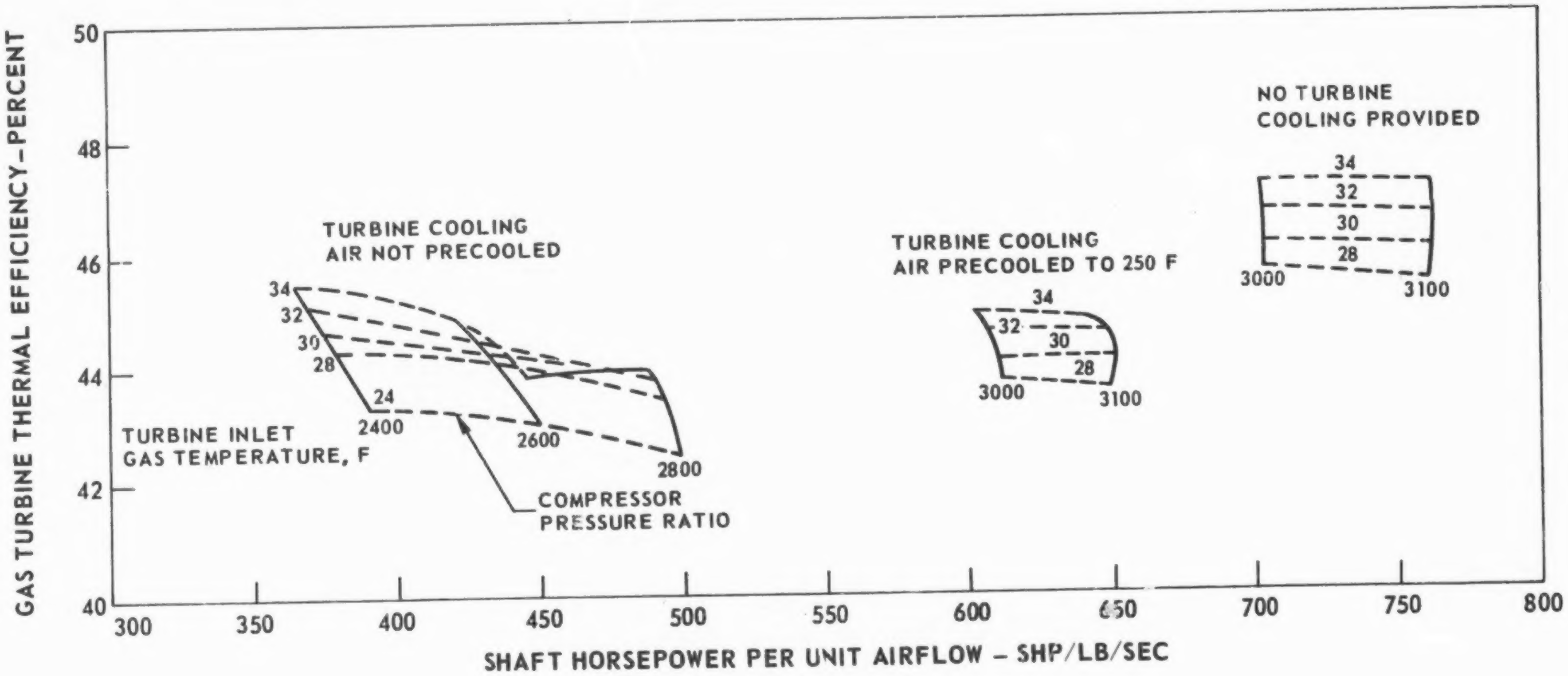


FIG. 45

FIG. 46

DEGRADATION IN SIMPLE - CYCLE GAS TURBINE THERMAL EFFICIENCY PERFORMANCE DUE TO TURBINE COOLING

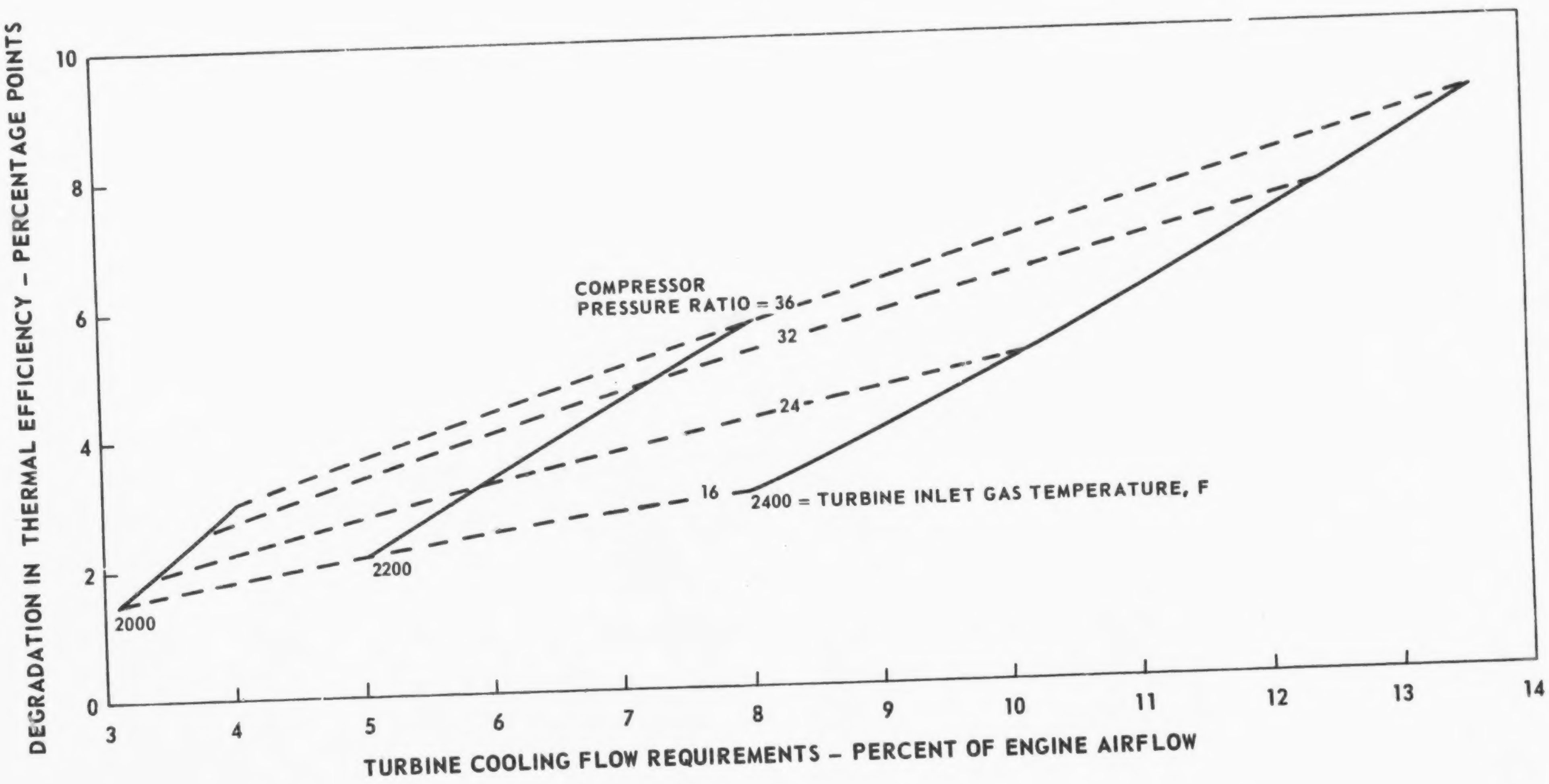
SECOND-GENERATION TECHNOLOGY

FUEL - GAS I (HHV = 162 BTU/FT^3)

AMBIENT - 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT - CONVECTION;

COMPRESSOR BLEED AIR UNCOOLED



DEGRADATION IN SIMPLE – CYCLE GAS TURBINE SPECIFIC OUTPUT PERFORMANCE DUE TO TURBINE COOLING

SECOND–GENERATION TECHNOLOGY

FUEL – GAS I (HHV = 162 BTU/FT^3)

AMBIENT – 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT – CONVECTION;
COMPRESSOR BLEED AIR UNCOOLED

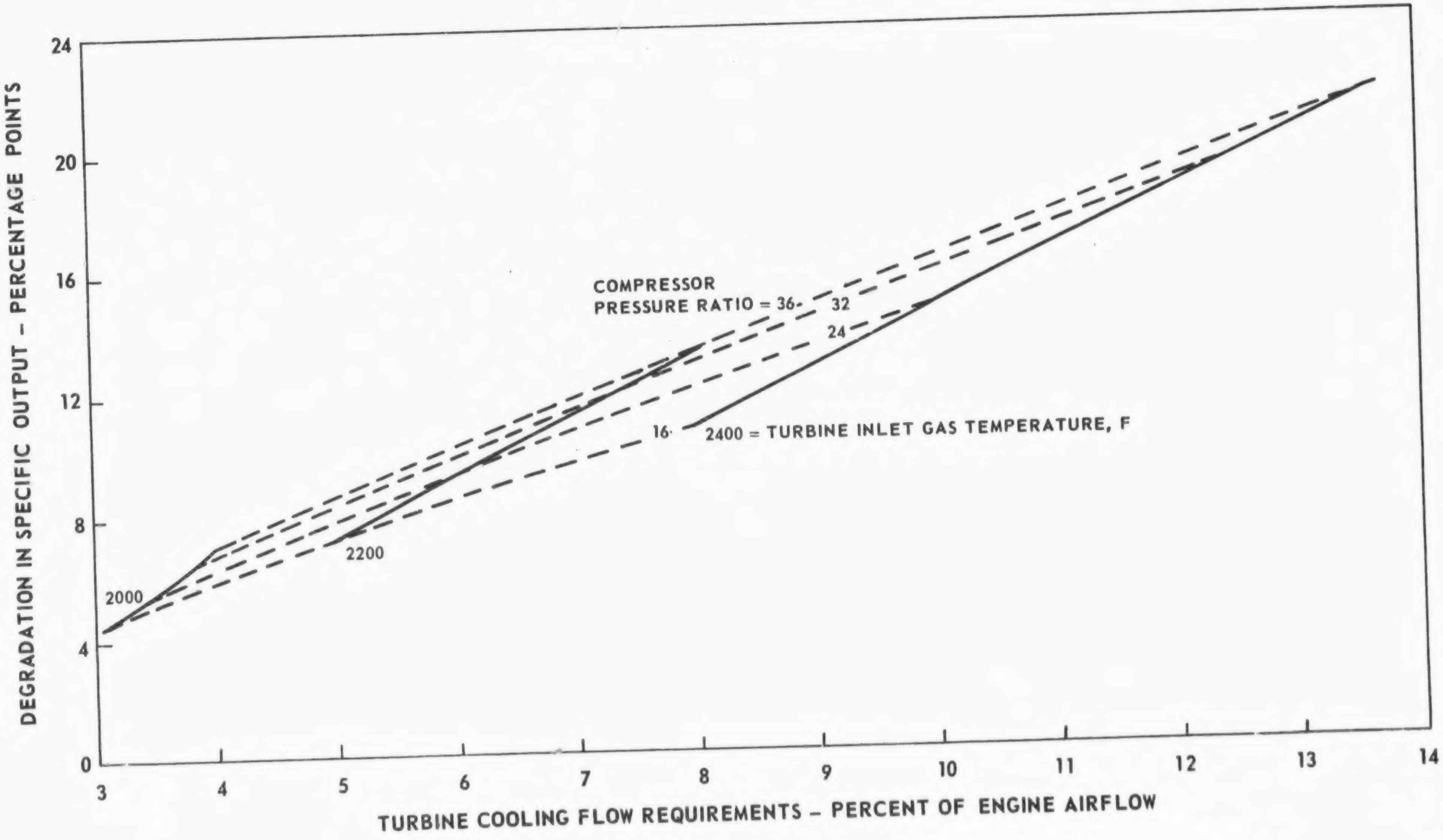
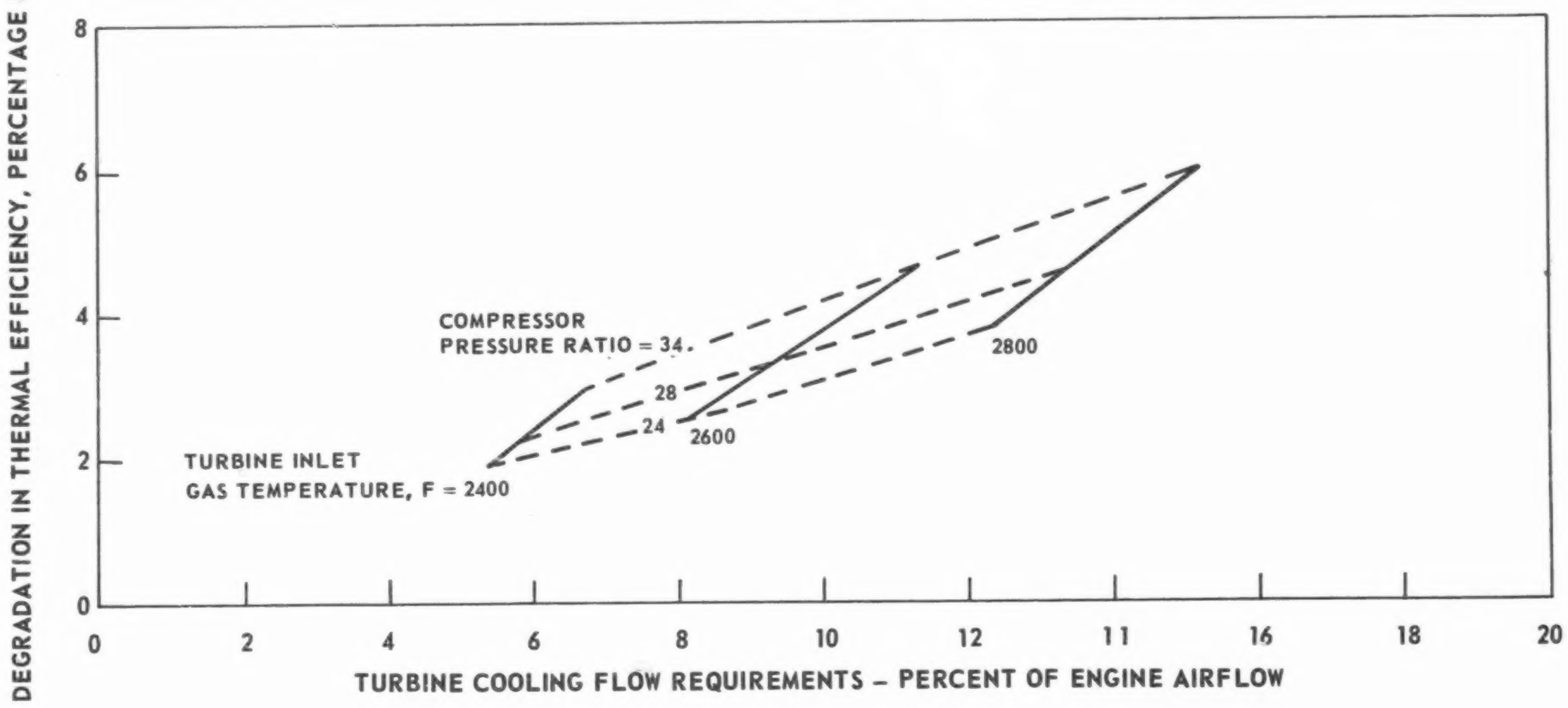


FIG. 47

FIG. 48

DEGRADATION IN SIMPLE – CYCLE GAS TURBINE THERMAL EFFICIENCY PERFORMANCE DUE TO TURBINE COOLING

THIRD-GENERATION TECHNOLOGY
FUEL – GAS I (HHV = 162 BTU/FT^3)
AMBIENT – 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT – CONVECTION;
COMPRESSOR BLEED AIR UNCOOLED



DEGRADATION IN SIMPLE – CYCLE GAS TURBINE SPECIFIC OUTPUT PERFORMANCE DUE TO TURBINE COOLING

THIRD–GENERATION TECHNOLOGY
FUEL – GAS I (HHV = 162 BTU/FT^3)
AMBIENT – 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT – CONVECTION; COMPRESSOR BLEED AIR UNCOOLED

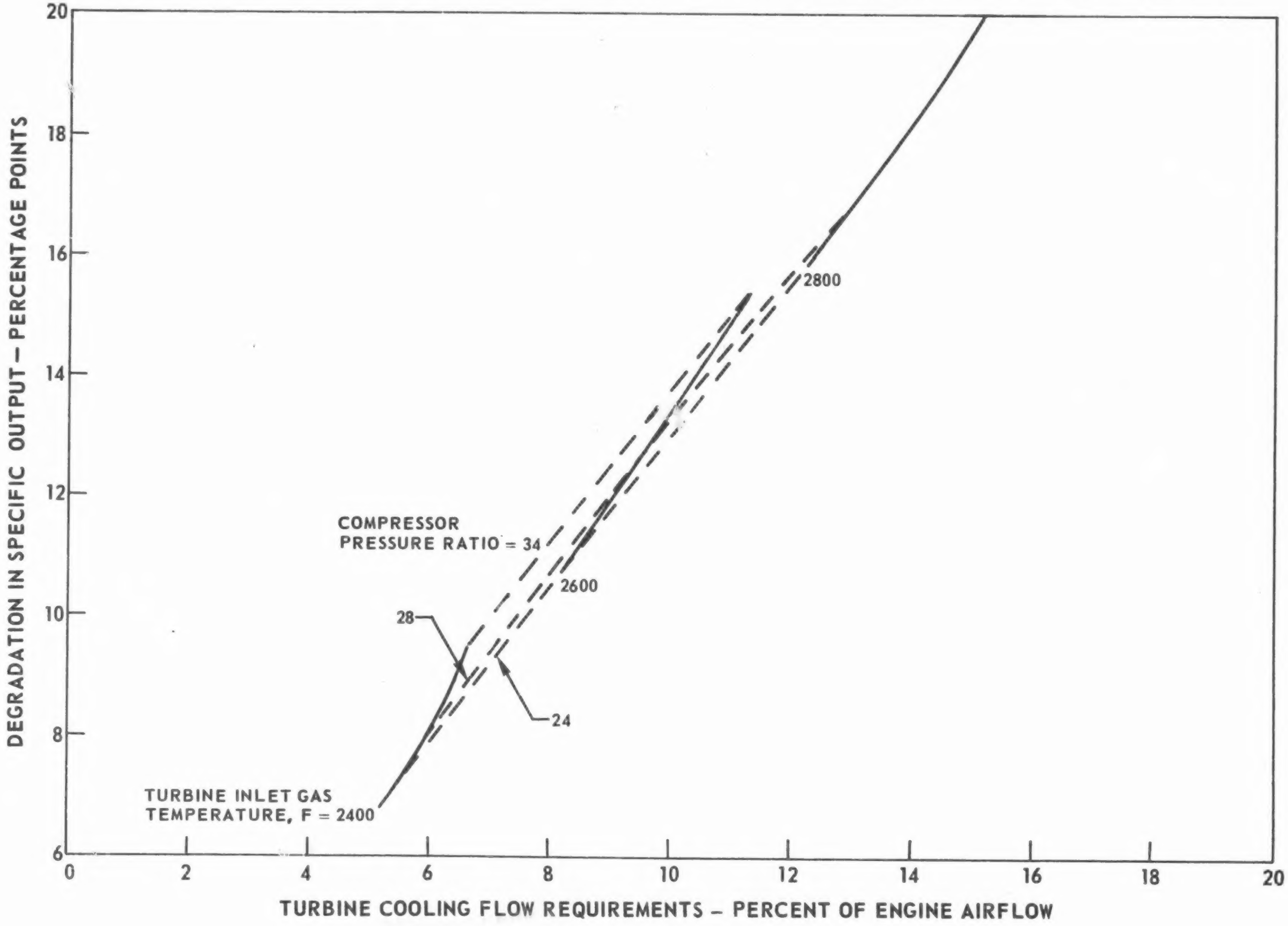
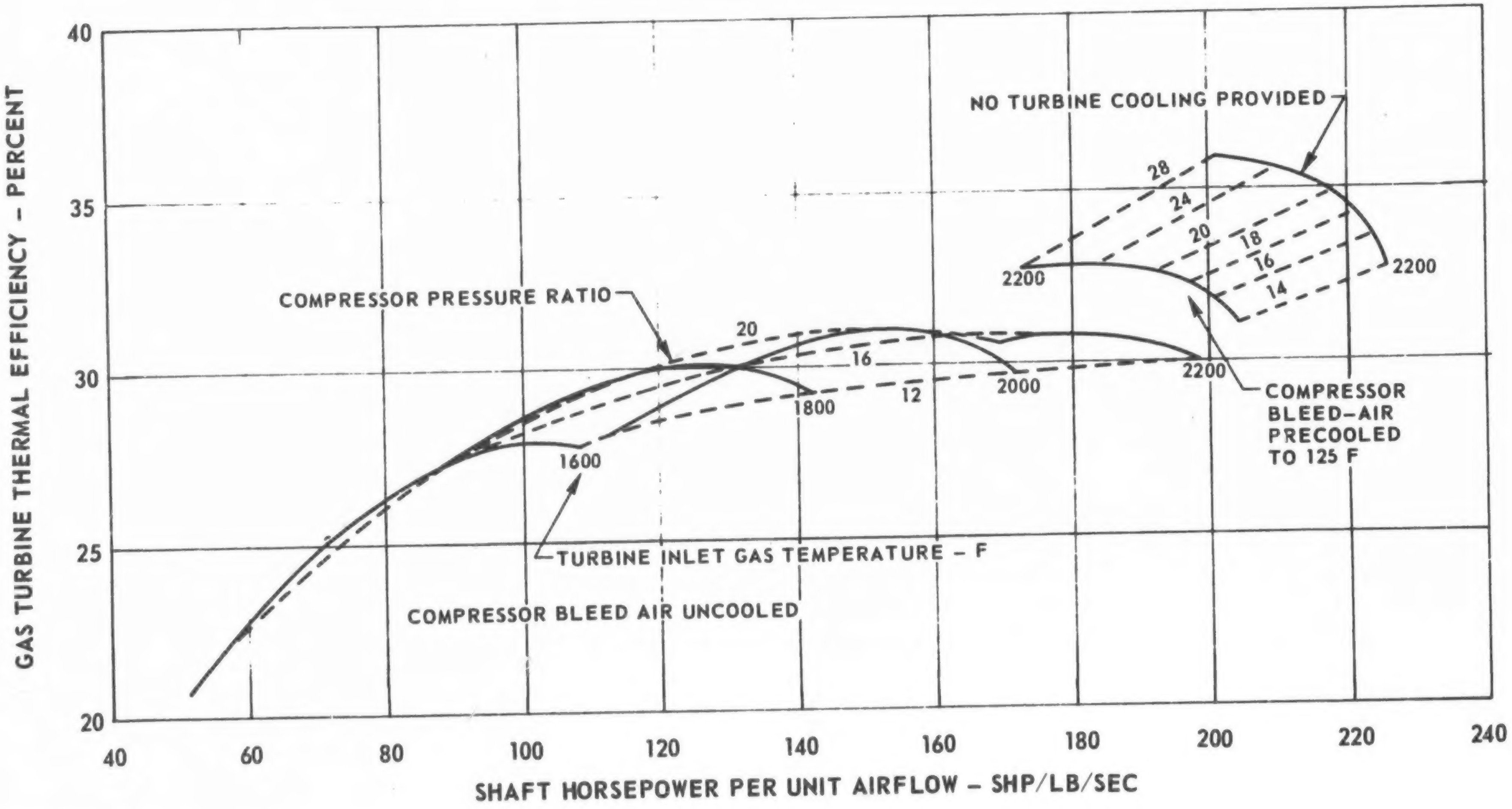


FIG. 49

FIG. 50

SIMPLE-CYCLE GAS TURBINE PERFORMANCE WITH HIGH HEAT – CONTENT FUELS

FIRST-GENERATION TECHNOLOGY
FUEL: METHANE (HHV = 1000 BTU/FT^3)
AMBIENT: 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION
ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER
REQUIREMENTS NOT INCLUDED



SIMPLE-CYCLE GAS TURBINE PERFORMANCES WITH HIGH HEAT-CONTENT FUELS

SECOND – GENERATION
FUEL: METHANE (HHV = 1000 BTU/FT^3)
AMBIENT: 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT–CONVECTION
ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS
NOT INCLUDED

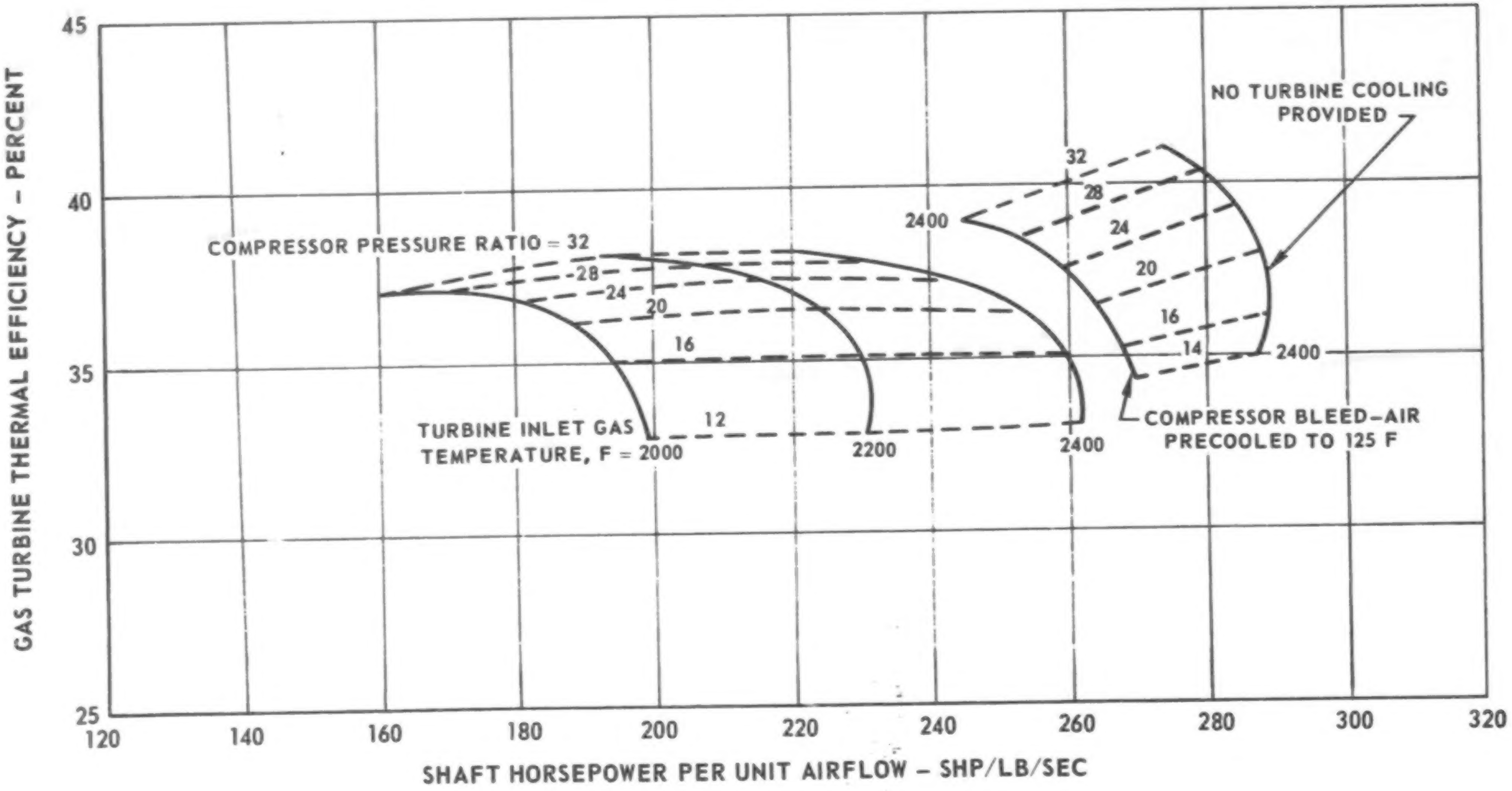


FIG. 51

FIG. 52

SIMPLE-CYCLE GAS TURBINE PERFORMANCE WITH HIGH-HEAT-CONTENT FUELS

THIRD-GENERATION TECHNOLOGY

FUEL: METHANE (HHV = 1000 BTU/FT^3)

AMBIENT: 80 F AND 1000 FT

ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

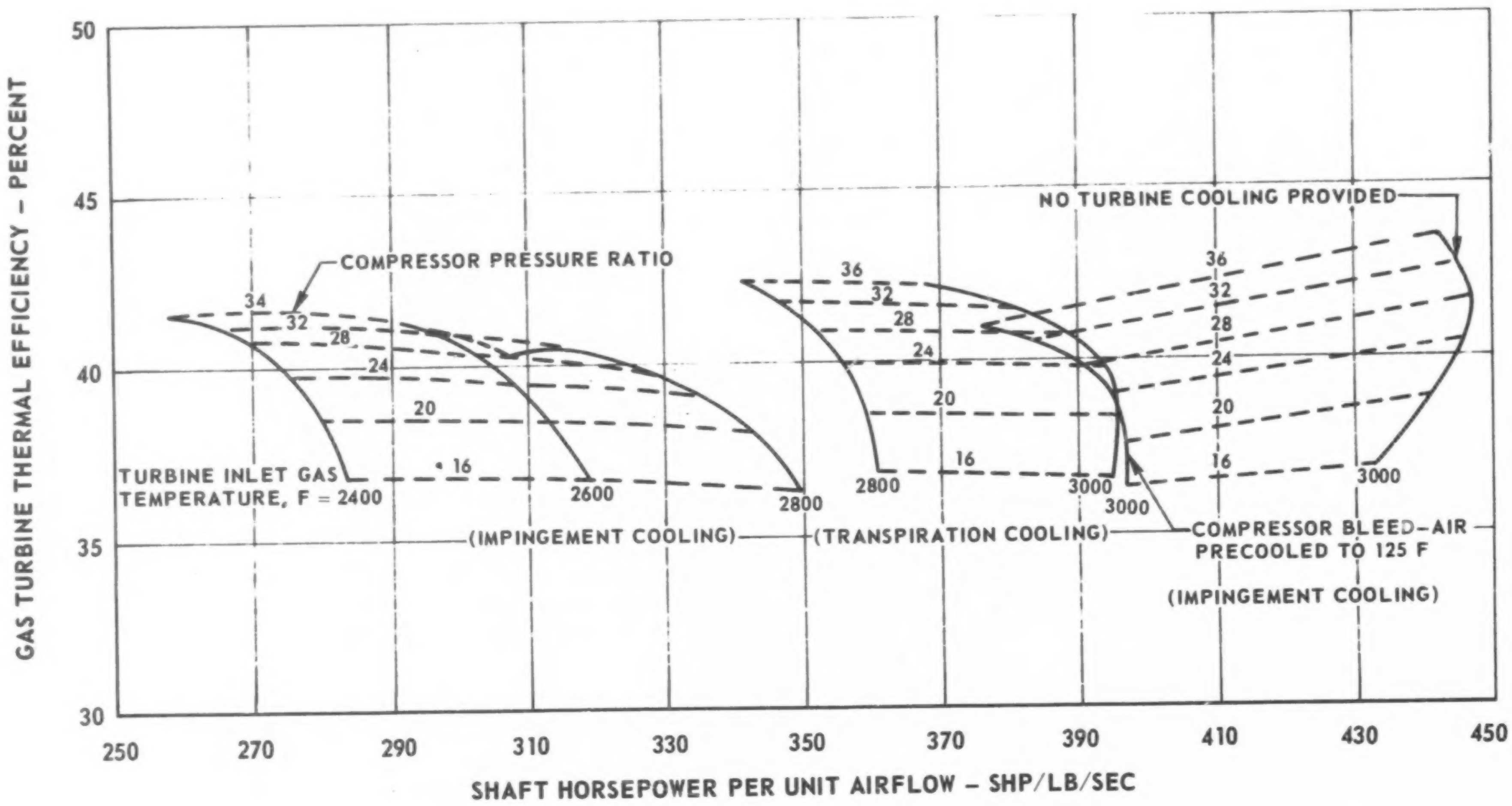
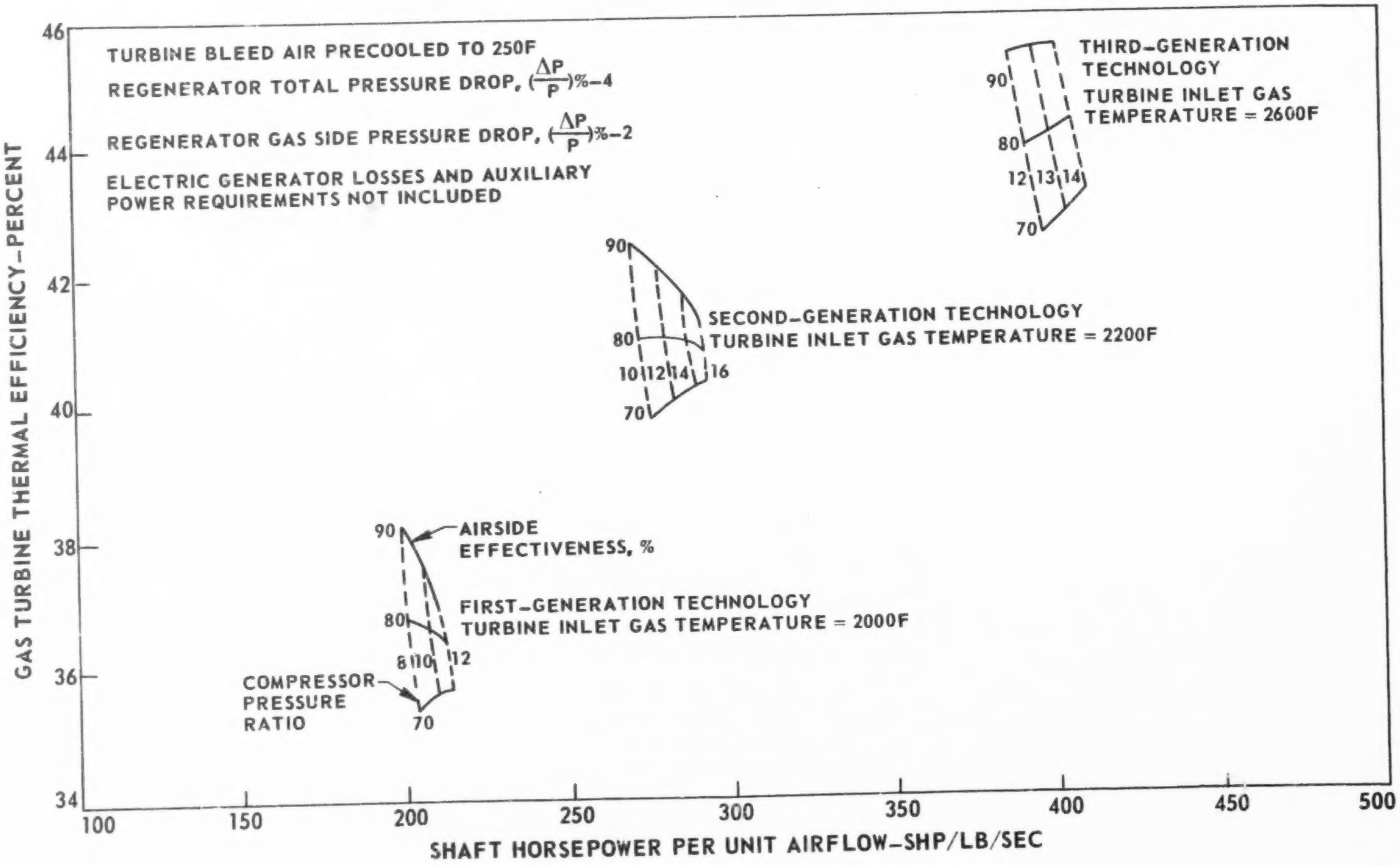


FIG. 53

BASE-LOAD REGENERATIVE-CYCLE GAS TURBINE PERFORMANCE WITH LOW-HEAT-CONTENT FUELS

FUEL- GAS I (HHV = 162 BTU/FT^3)
AMBIENT-80F AND 1000FT



TURBINE BLEED AIR PRECOOLED TO 250F

REGENERATOR TOTAL PRESSURE DROP, $(\frac{\Delta P}{P})$%-4

REGENERATOR GAS SIDE PRESSURE DROP, $(\frac{\Delta P}{P})$%-2

ELECTRIC GENERATOR LOSSES AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

FIG. 54

BASE-LOAD REGENERATIVE-CYCLE GAS TURBINE PERFORMANCE WITH HIGH HEAT – CONTENT FUELS

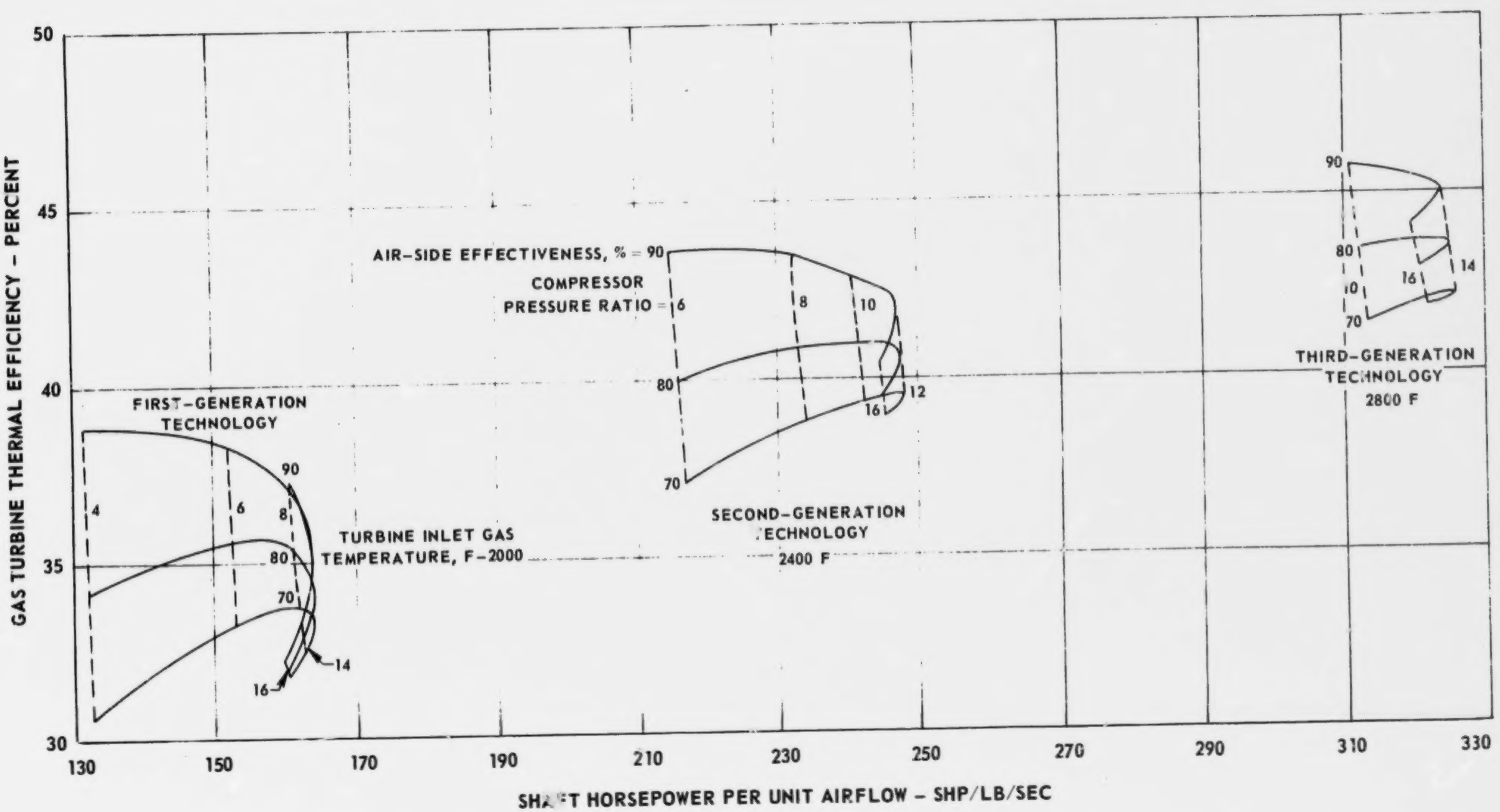
FUEL: METHANE (HHV = 1000 BTU/FT^3)

AMBIENT: 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION

ELECTRIC GENERATOR EFFICIENCY AND AUXILIARY POWER REQUIREMENT EFFECTS EXCLUDED

REGENERATOR TOTAL PRESSURE DROP, $\frac{\Delta P}{P} = 4\%$



REGENERATOR GAS TEMPERATURES FOR FIRST-GENERATION TECHNOLOGY DESIGNS

FUEL: METHANE (HHV = 1000 BTU/FT^3)
AMBIENT: 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION
REGENERATOR PRESSURE LOSS, $\frac{\Delta P}{P}$ = 4.0%
REGENERATOR AIRSIDE EFFECTIVENESS = 80%
TURBINE INLET GAS TEMPERATURE, F = 2400

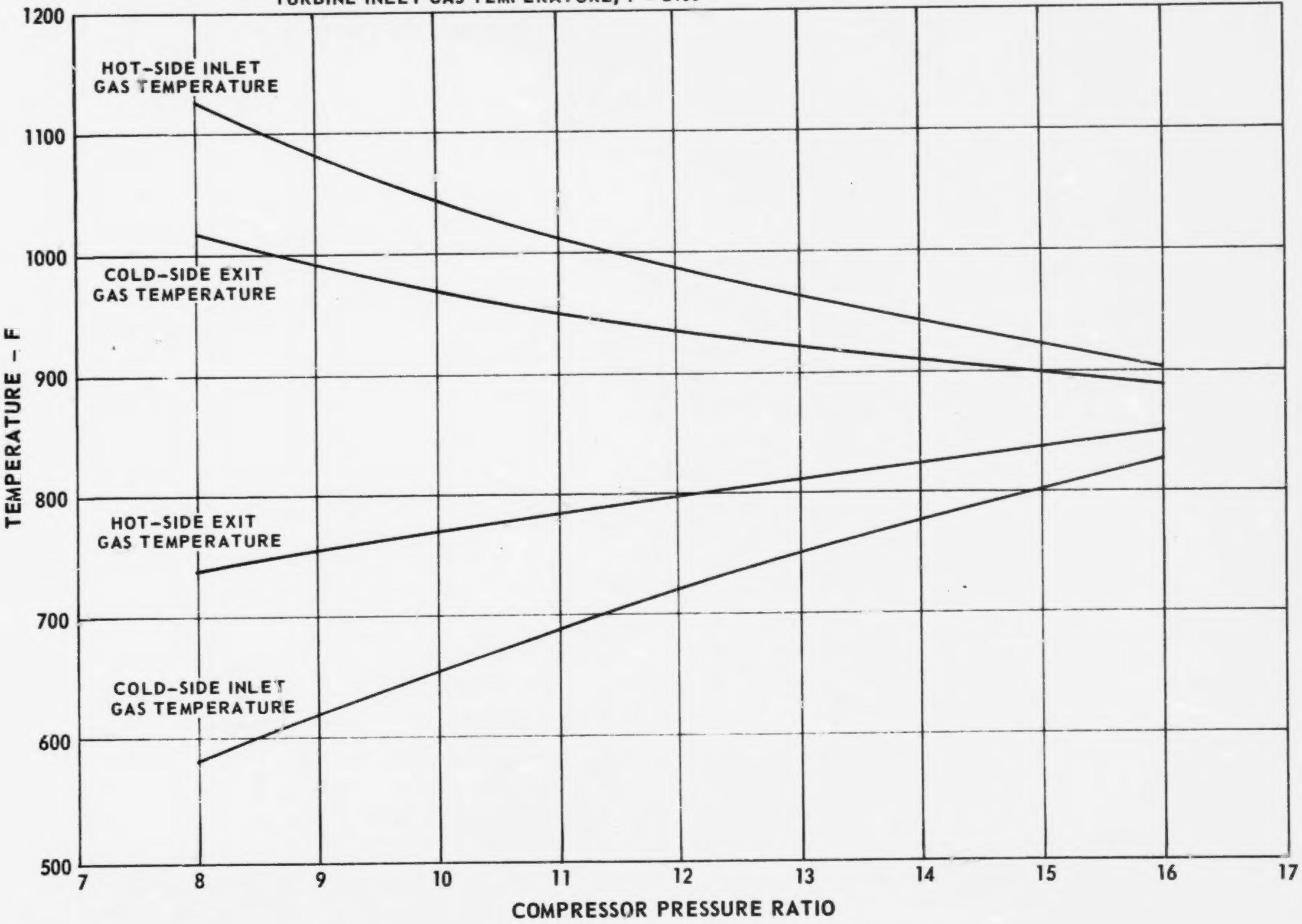
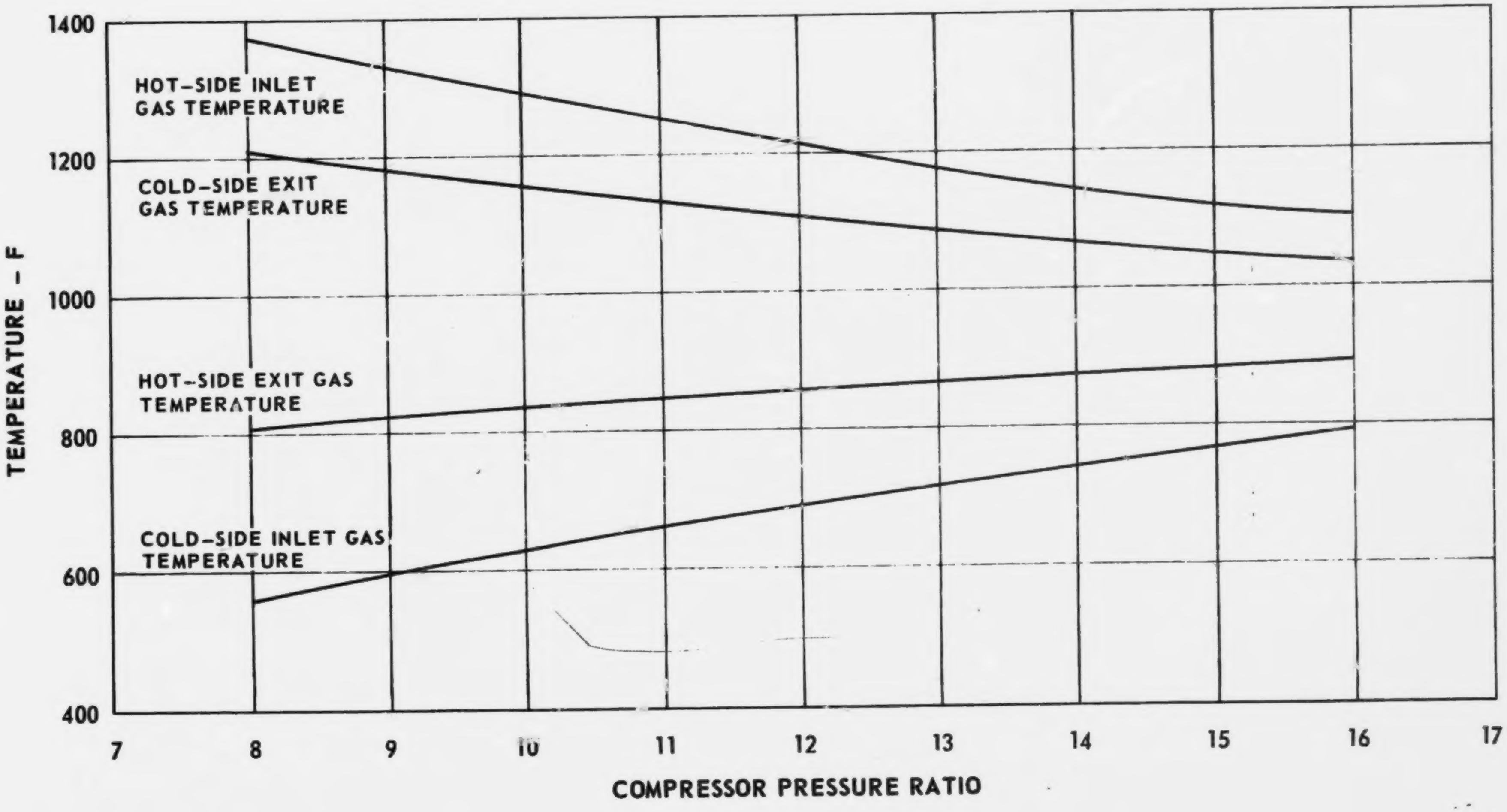


FIG. 55

FIG. 56

REGENERATOR GAS TEMPERATURES FOR SECOND-GENERATION TECHNOLOGY DESIGNS

FUEL: METHANE (HHV = 1000 BTU/FT^3)
AMBIENT: 80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION
REGENERATOR PRESSURE LOSS, $\frac{\Delta P}{P}$ = 4.0%
REGENERATOR AIRSIDE EFFECTIVENESS = 80%
TURBINE INLET GAS TEMPERATURE, F = 2400



REGENERATOR GAS TEMPERATURES FOR THIRD-GENERATION TECHNOLOGY DESIGNS

FUEL: METHANE (HHV = 1000 BTU/FT^3)

AMBIENT: 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION

REGENERATOR PRESSURE LOSS, $\frac{\Delta P}{P}$ = 4.0%

REGENERATOR AIRSIDE EFFECTIVENESS = 80%

TURBINE INLET GAS TEMPERATURE, F = 2800

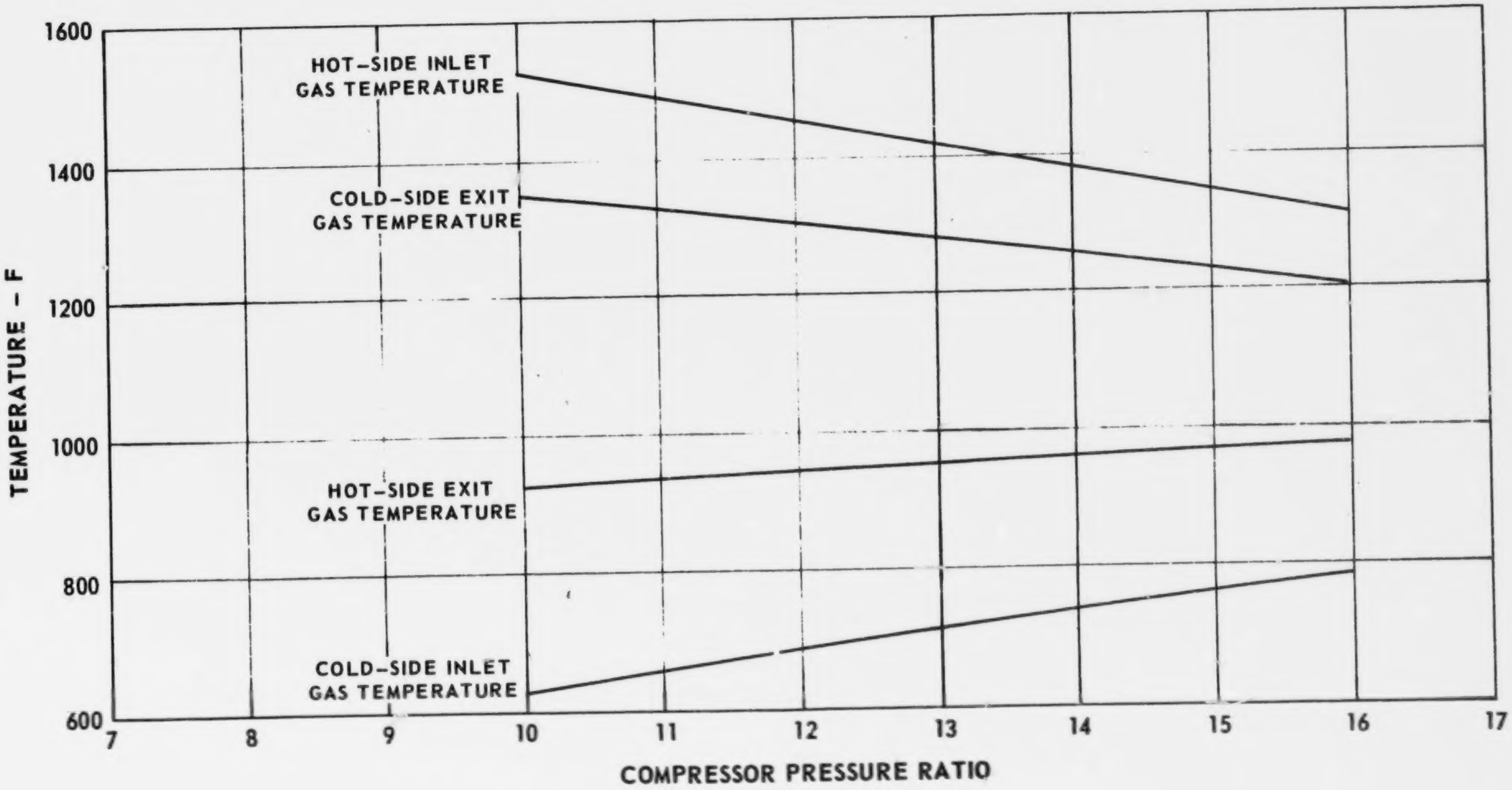
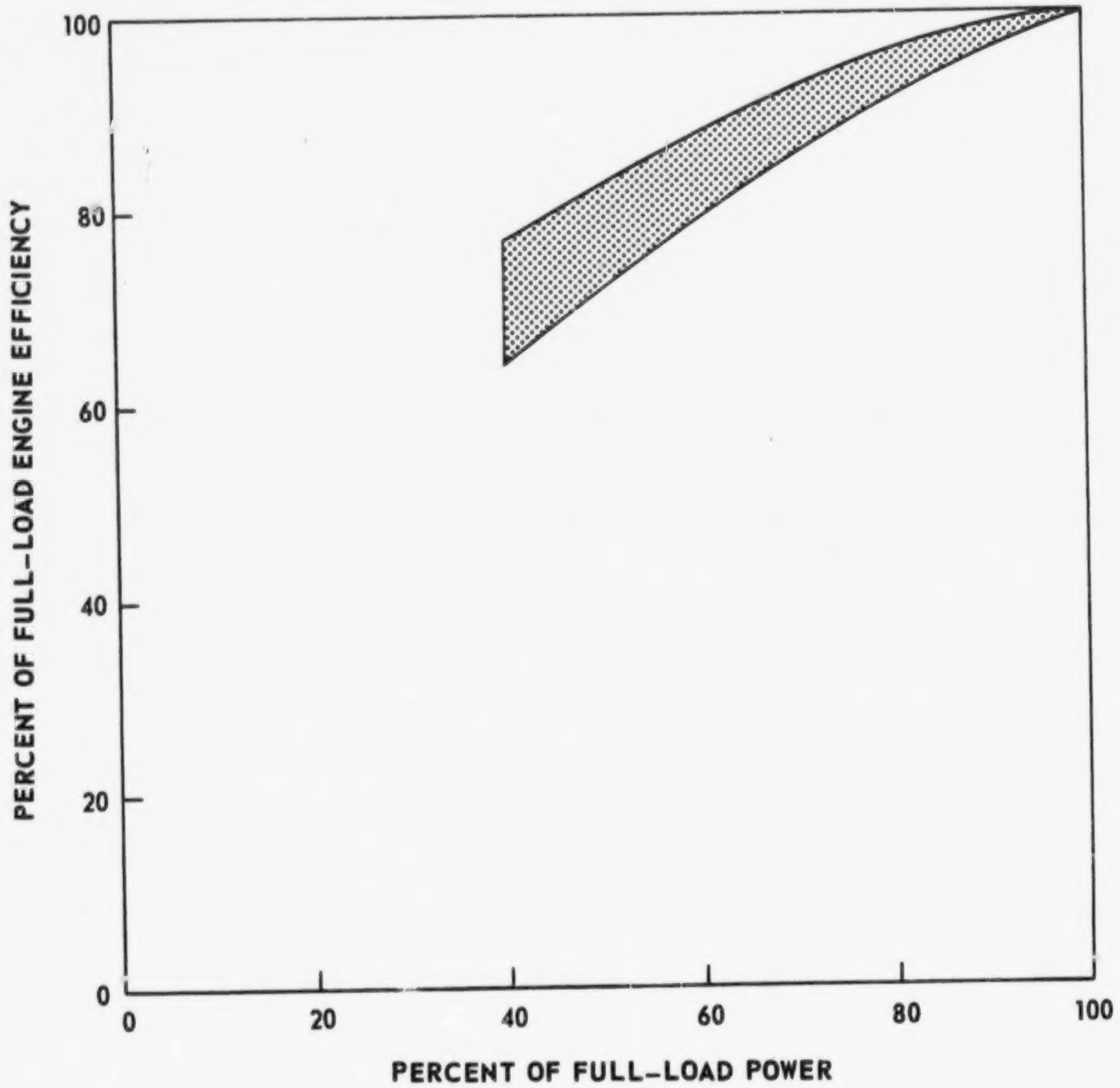


FIG. 57.

FIG. 58

PART-LOAD PERFORMANCE FOR TWO-SHAFT GAS TURBINE ENGINES

DATA FROM REF. 95



COMPARISON OF BUSBAR POWER COSTS FOR BASE-LOAD GAS TURBINE SYSTEMS

1000-MW NOMINAL SIZE

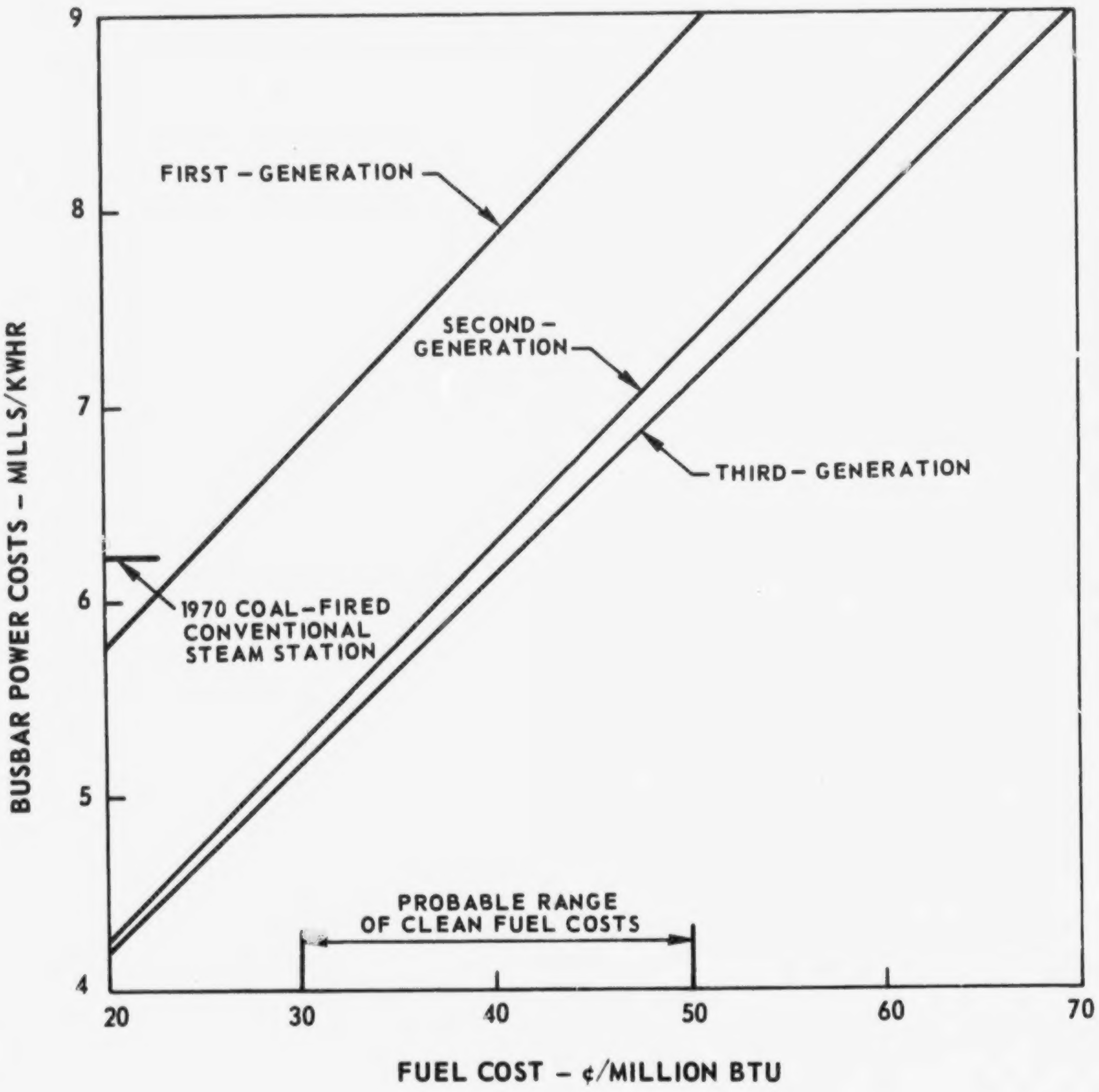


FIG. 60

EXHAUST-FIRED COGAS SYSTEM

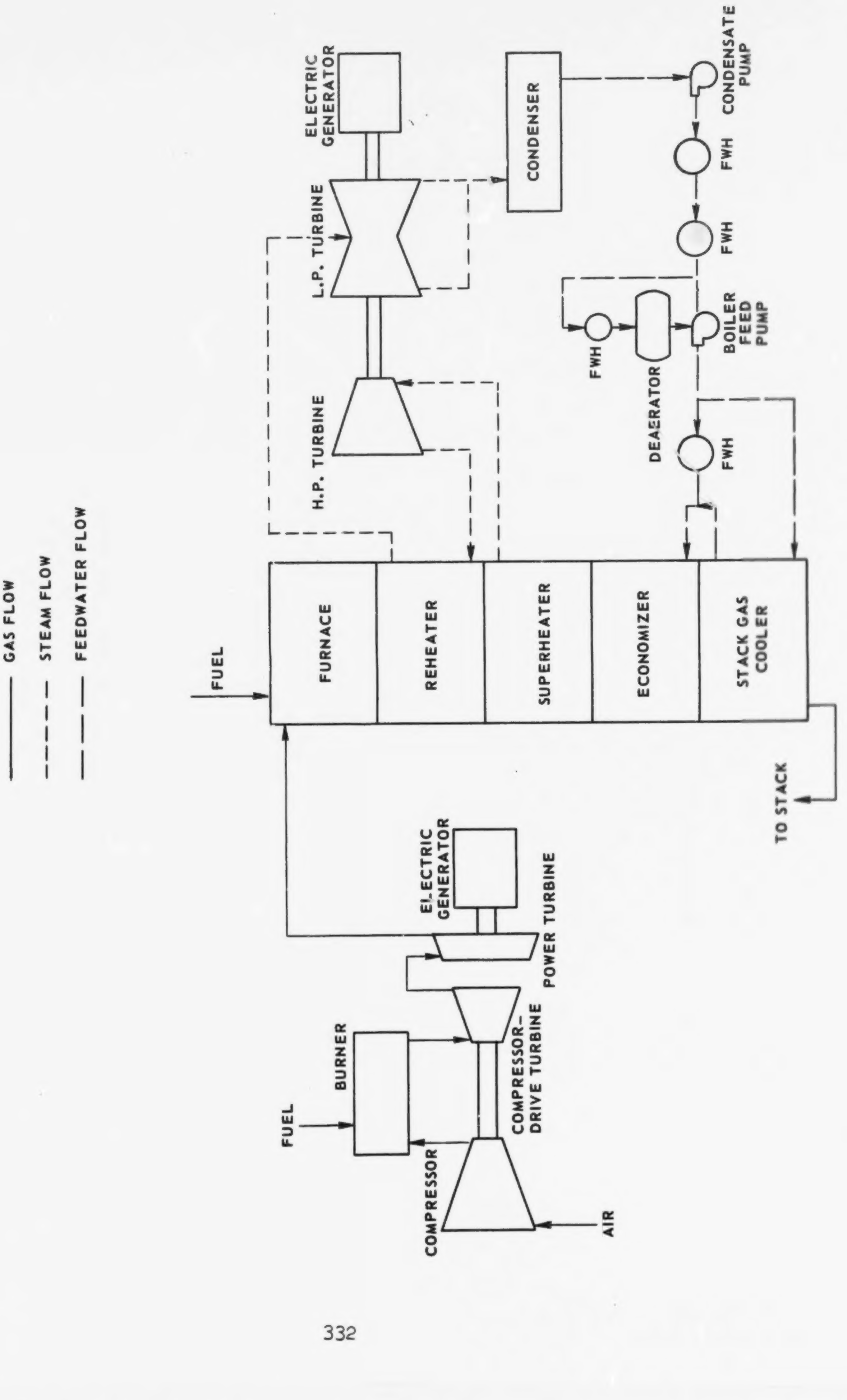


FIG. 61

WASTE-HEAT RECOVERY COGAS SYSTEM

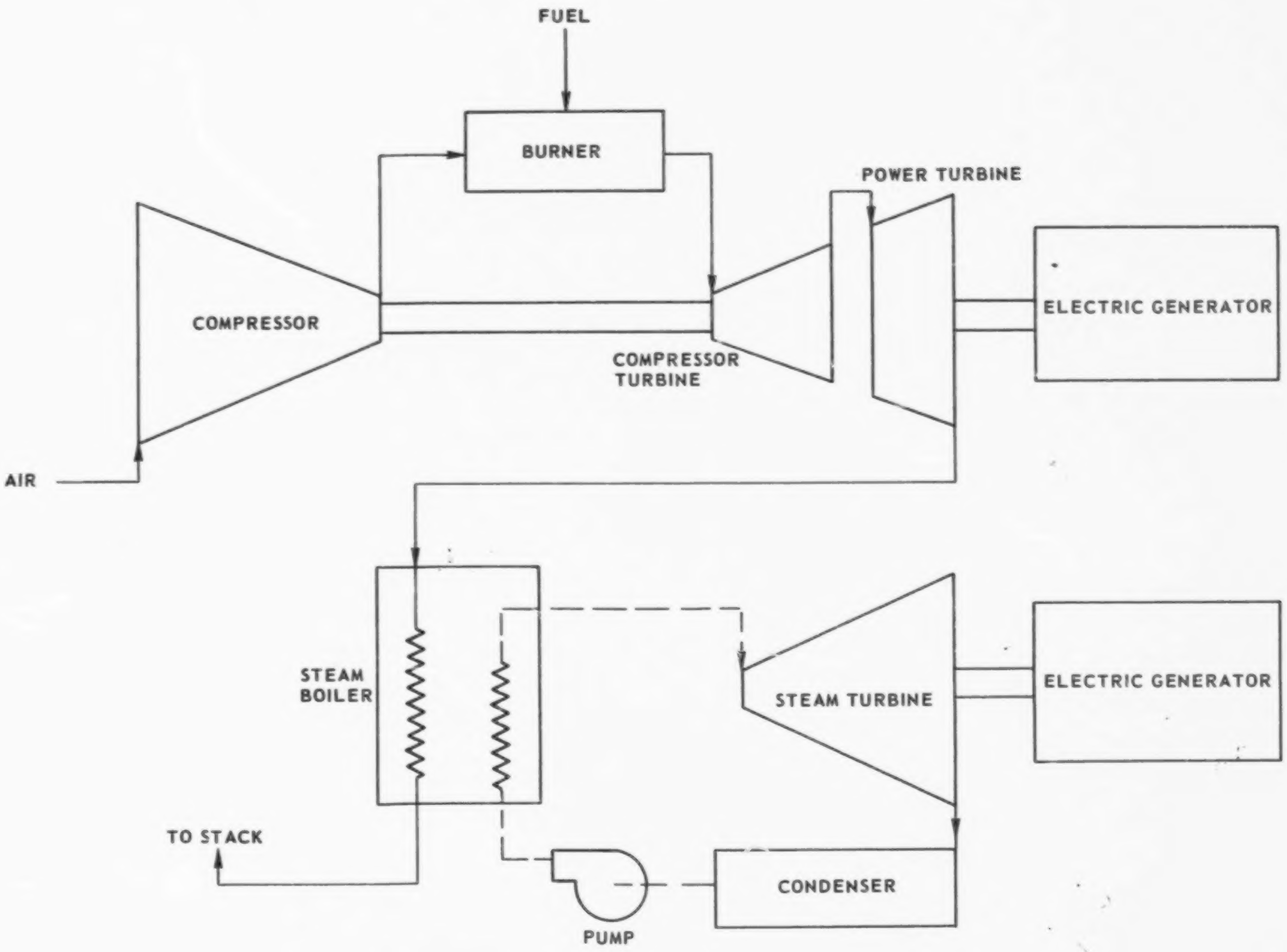


FIG. 62

CONVENTIONAL SUPERCHARGED COGAS SYSTEM

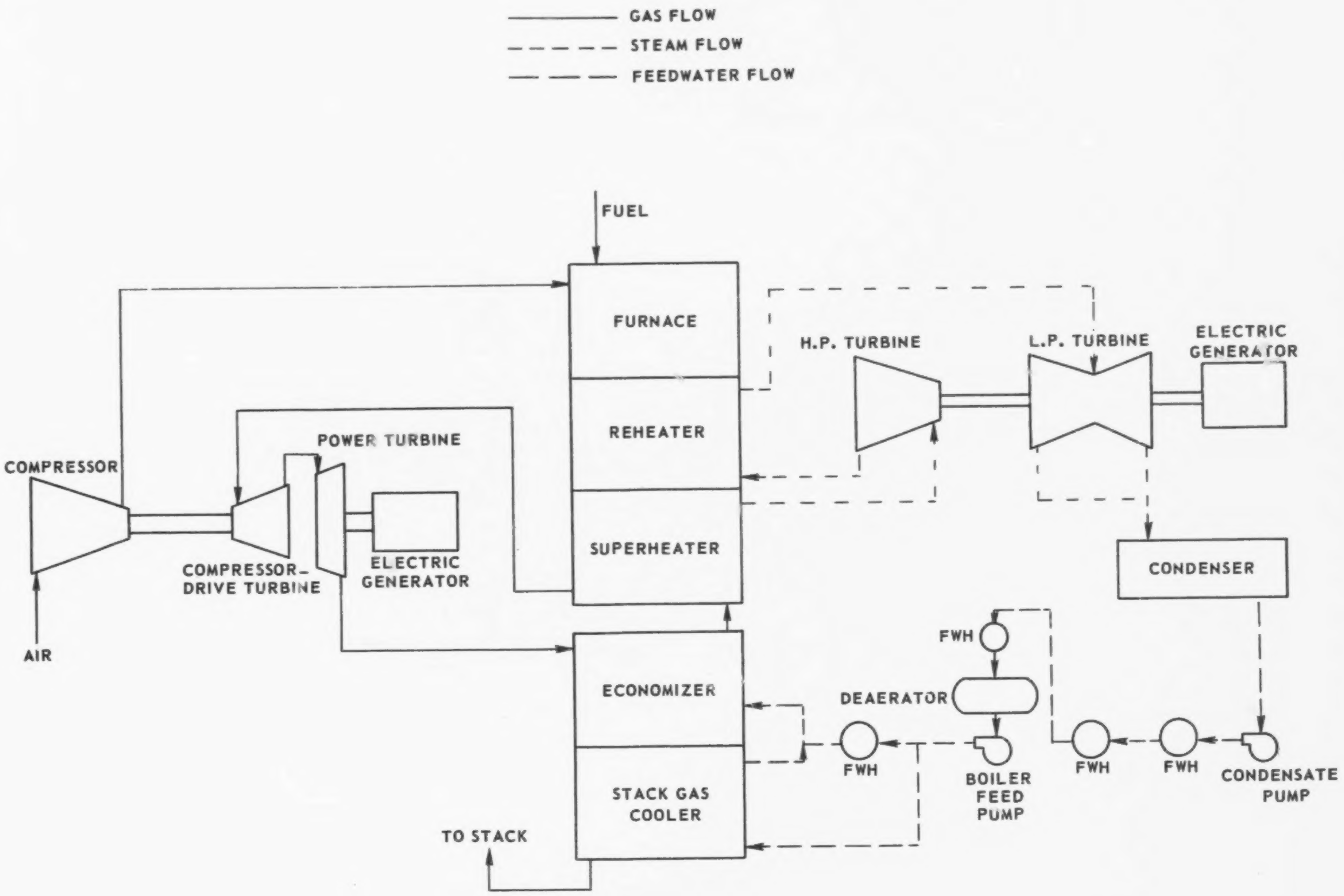


FIG. 63

EFFECT OF SUPERCHARGING ON BOILER HEAT TRANSFER AREA

DATA FROM REF. 122

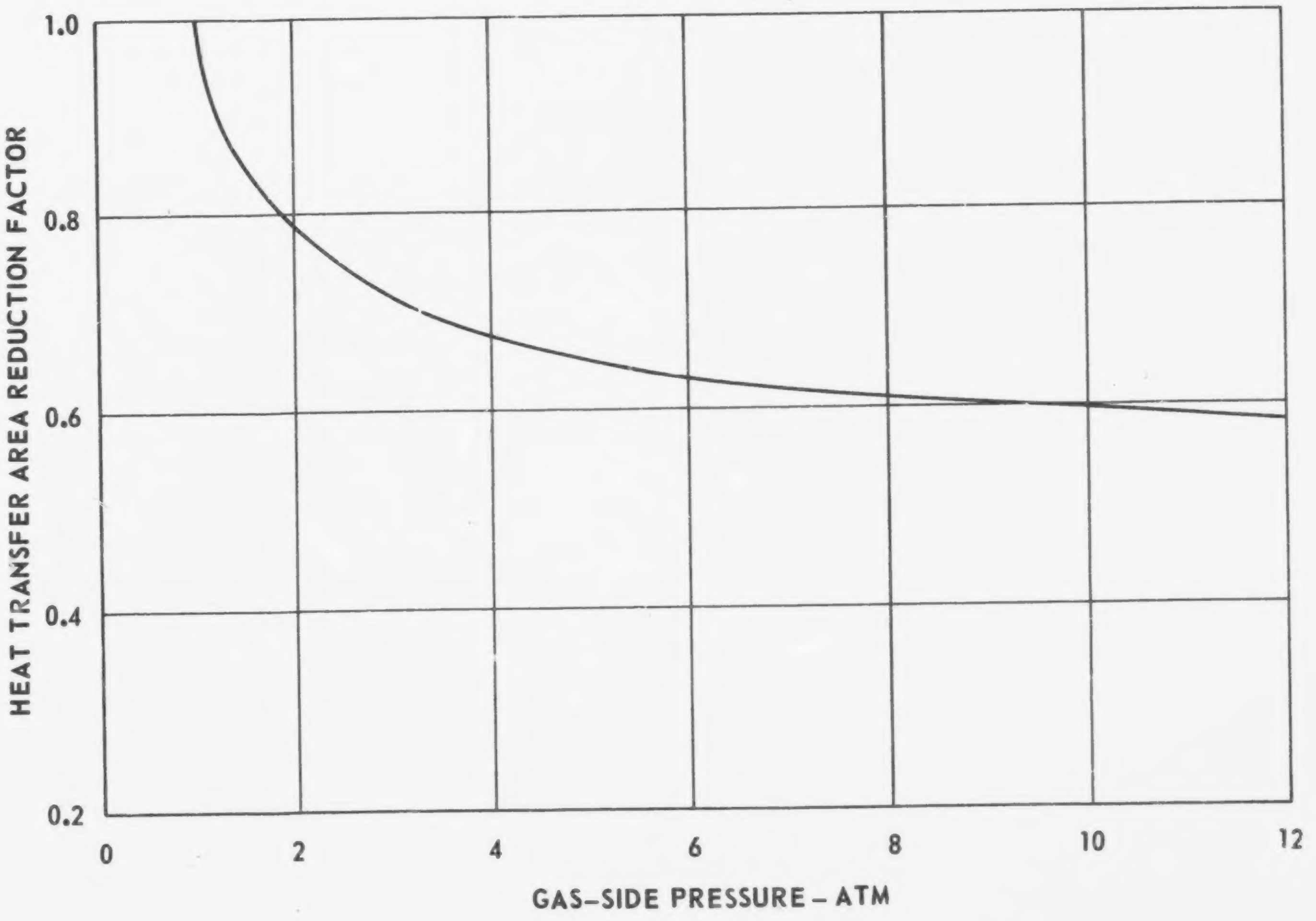
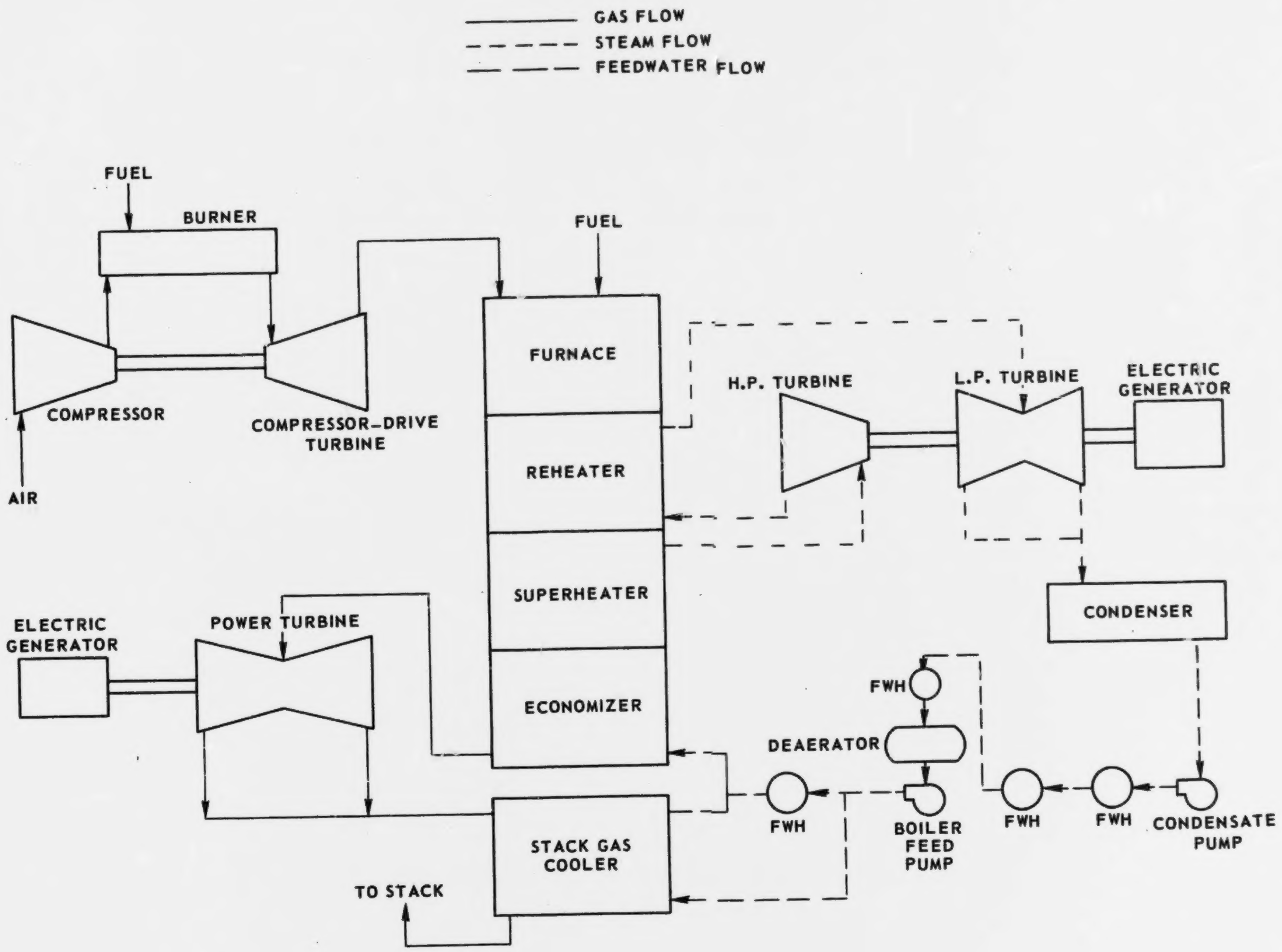


FIG. 64

GAS GENERATOR SUPERCHARGED COGAS SYSTEM



TWO-PRESSURE SUPERCHARGED COGAS SYSTEM

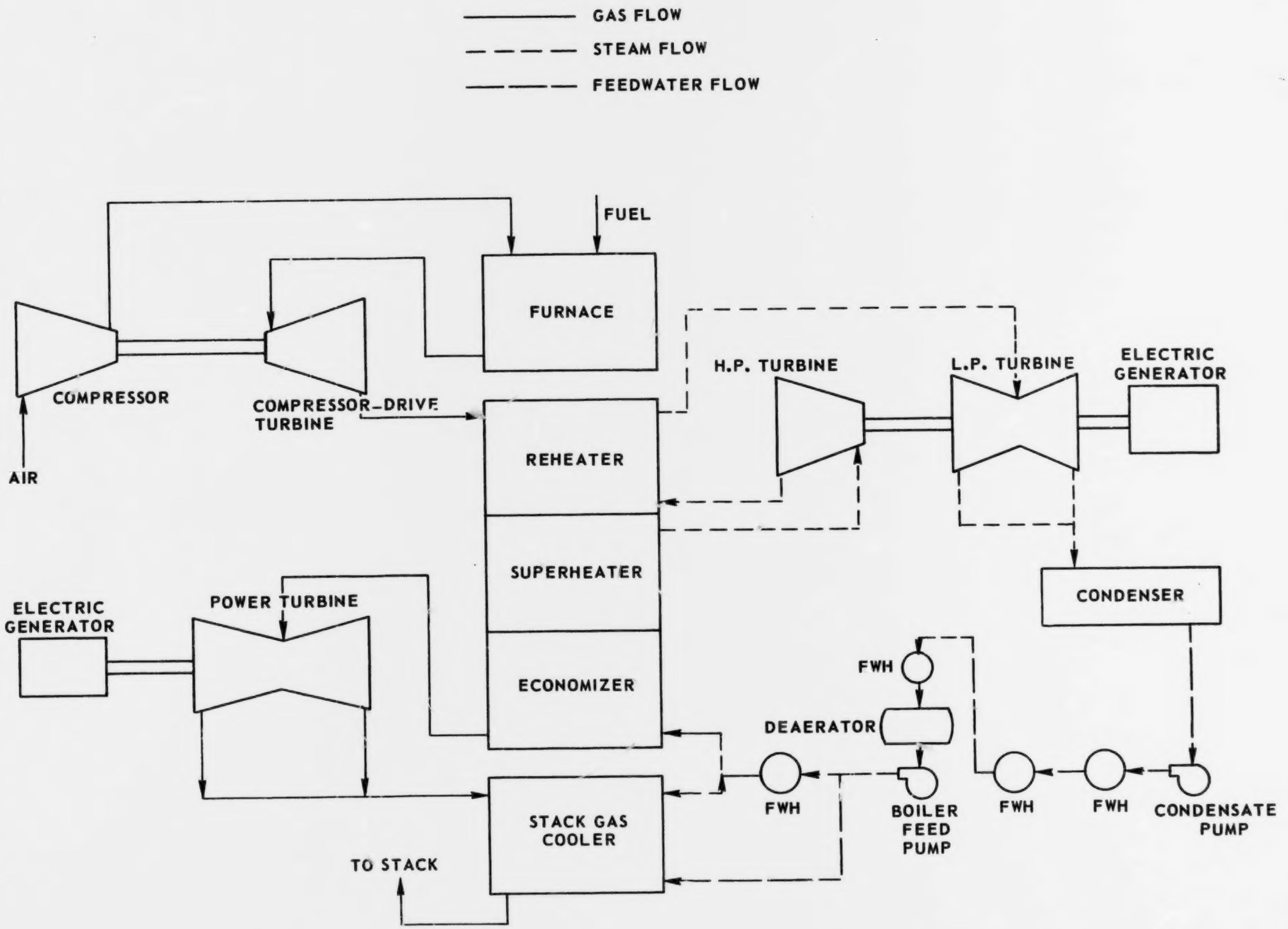
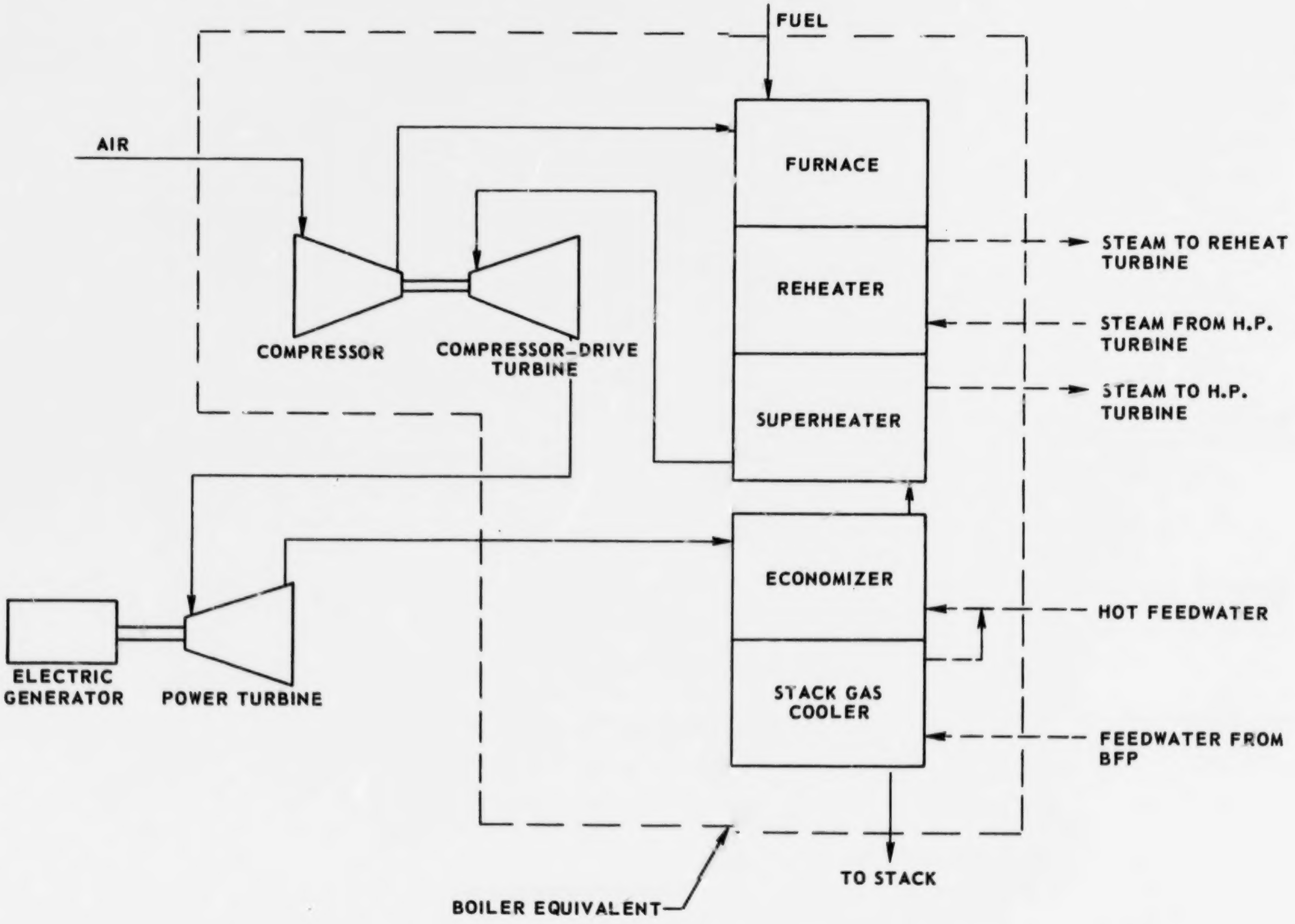


FIG. 65

FIG. 66

CONVENTIONAL SUPERCHARGED COGAS SYSTEM BOILER EQUIVALENT



COMPARATIVE PERFORMANCE OF COGAS SYSTEMS

FUEL – METHANE
COMPRESSOR PRESSURE RATIO = 12
TURBINE INLET TEMPERATURE = 2400 F
STEAM CYCLE EFFICIENCY = 40%
STACK GAS TEMPERATURE = 300 F

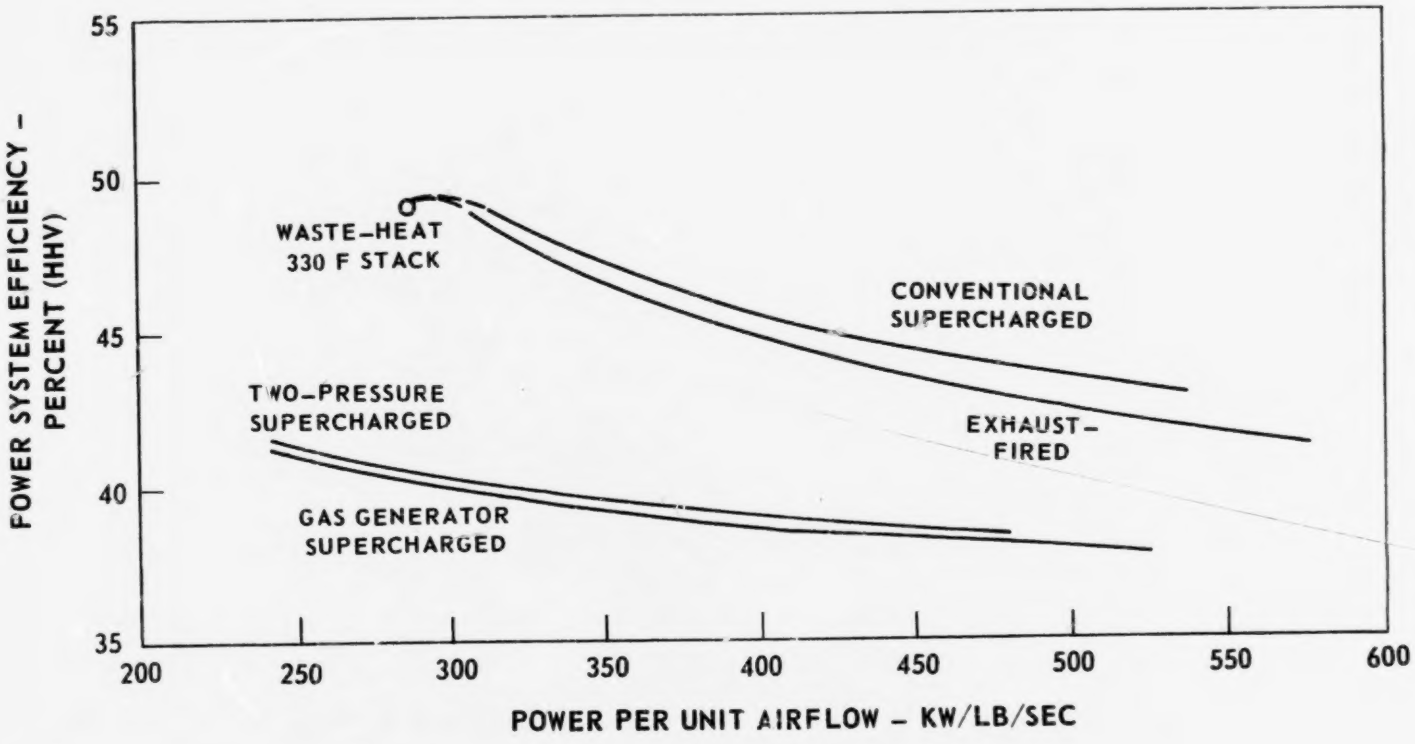


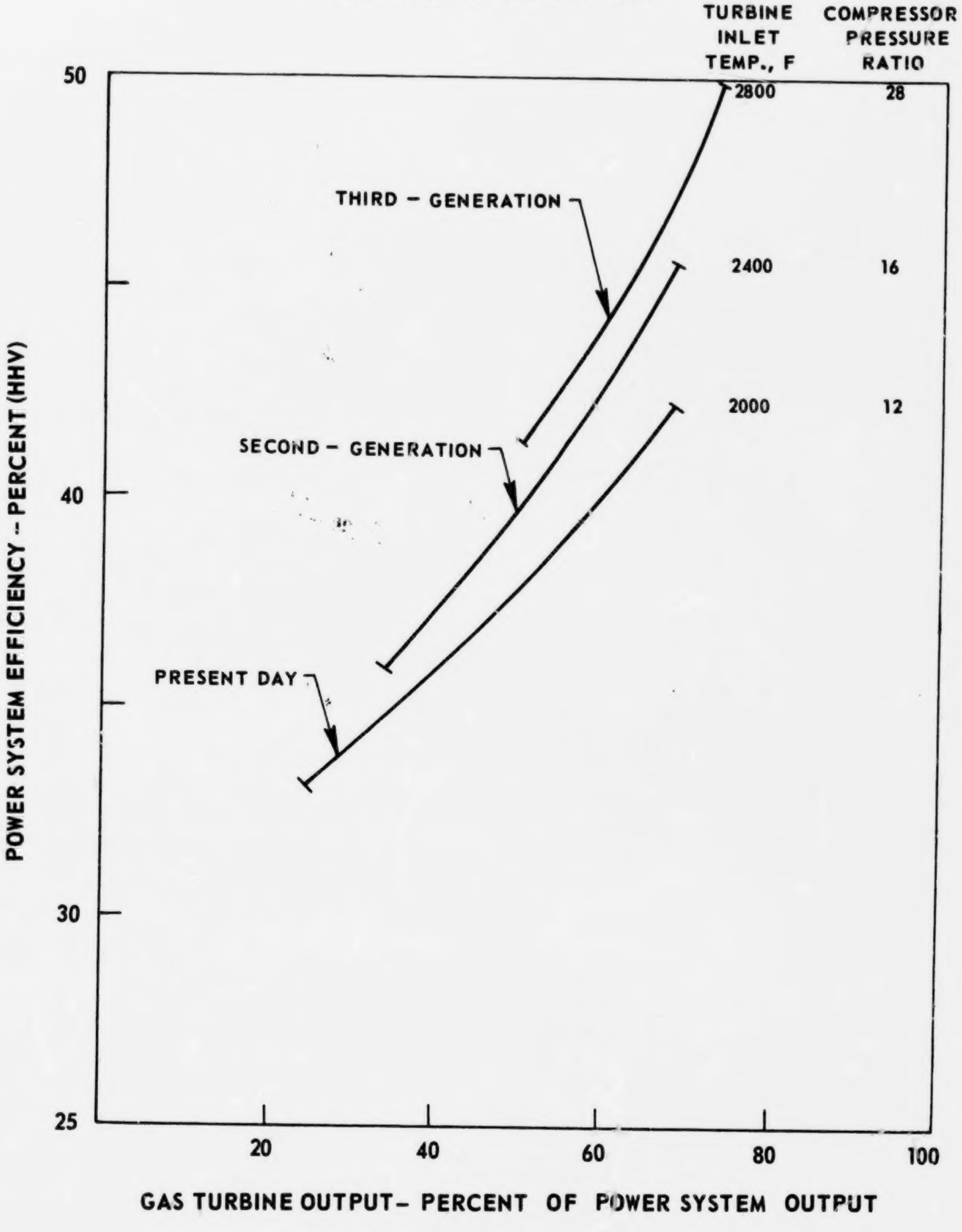
FIG. 67

FIG. 68

PERFORMANCE OF EXHAUST-FIRED COGAS SYSTEM

FUEL – METHANE

STEAM CYCLE EFFICIENCY = 34%



EFFECT OF FUEL ON PERFORMANCE OF WASTE – HEAT COGAS POWER SYSTEM

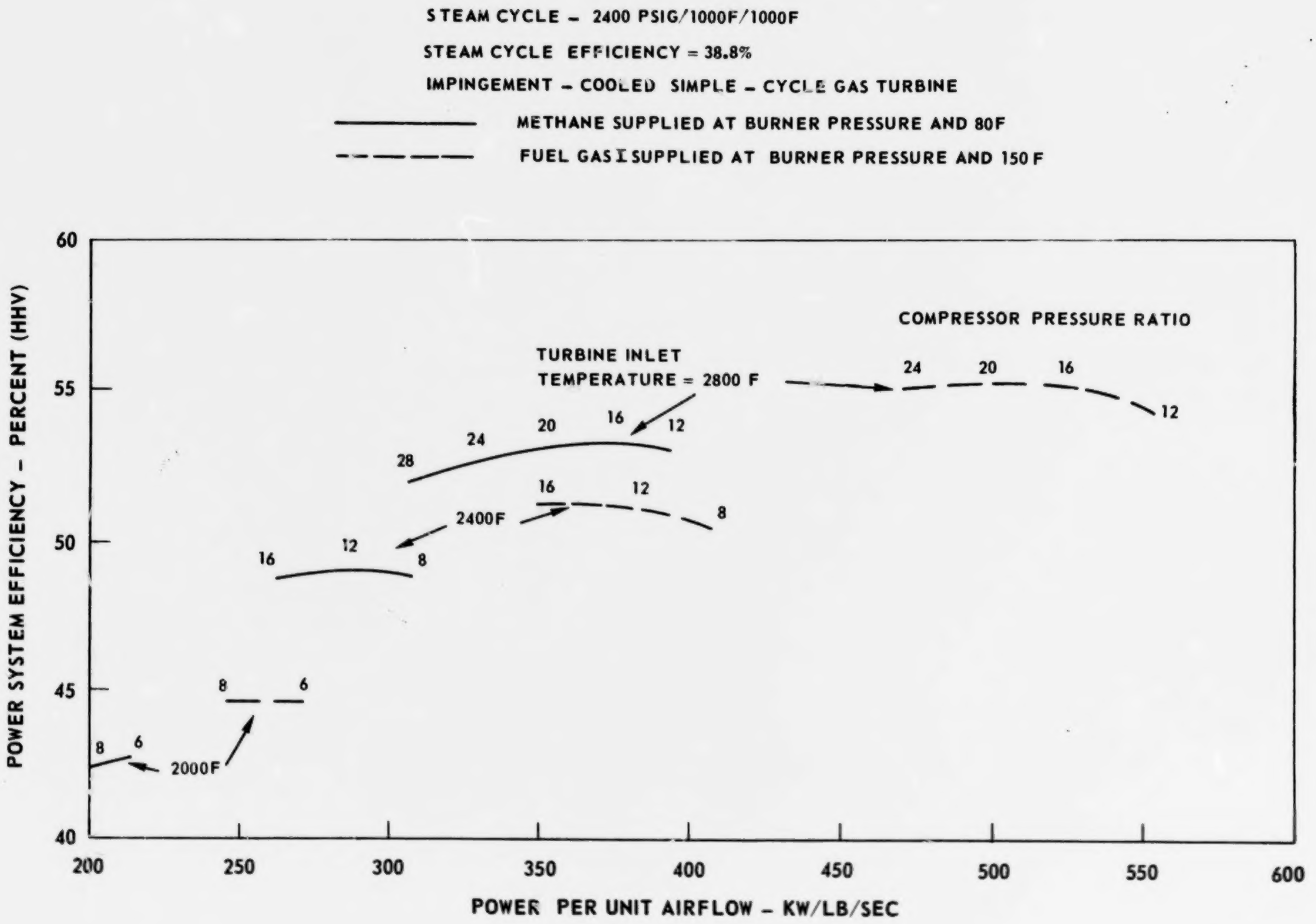
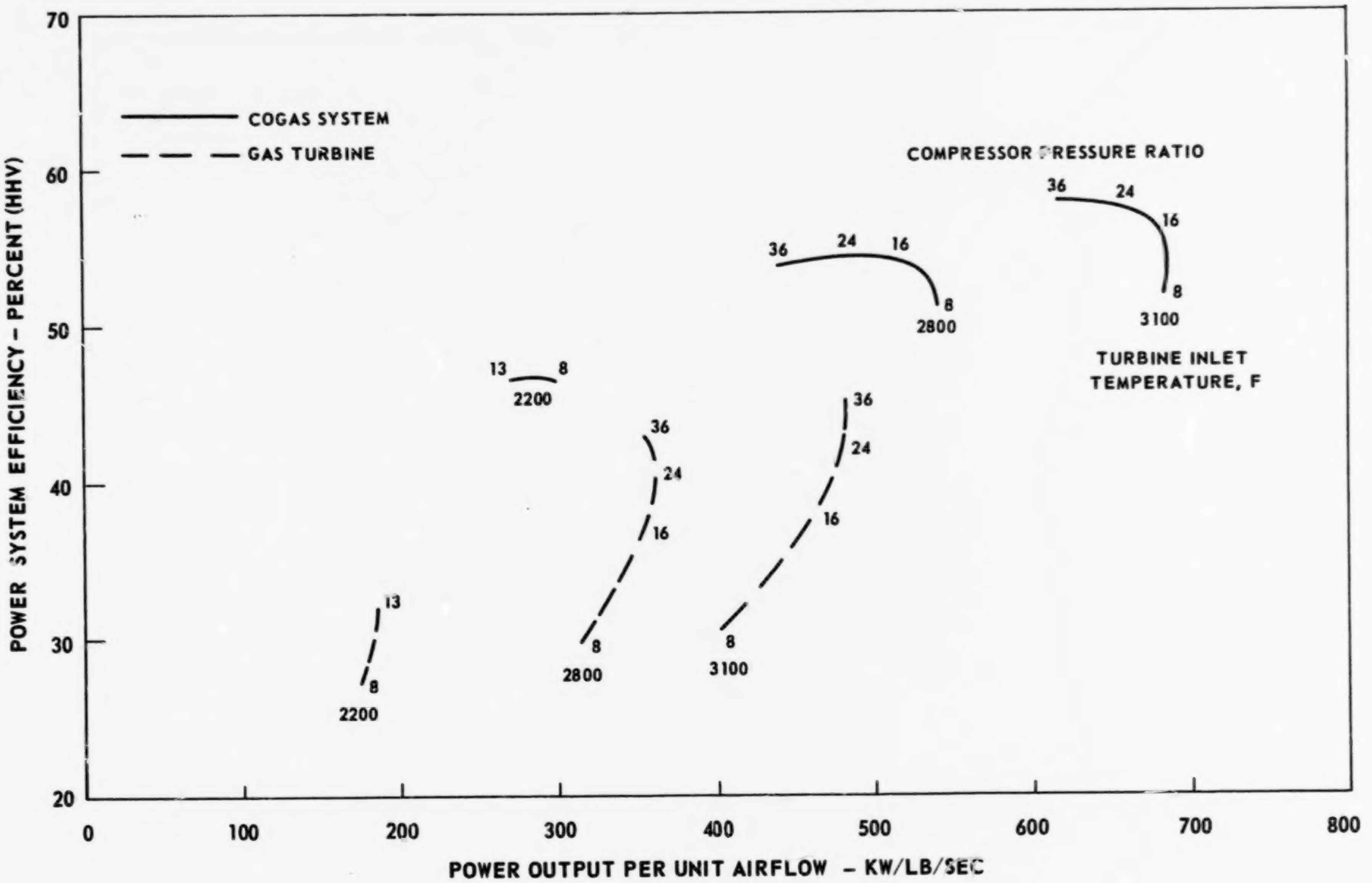


FIG. 69

FIG. 70

PERFORMANCE OF WASTE-HEAT COGAS POWER SYSTEM

FUEL GASI GAS SUPPLIED AT BURNER PRESSURE AND 150 F
STEAM CYCLE – 2400 PSIG/1000 F/1000 F
STEAM CYCLE EFFICIENCY = 38.8%
GAS TURBINE COOLING AIR PRECOOLED TO 125 F
GAS TURBINE BACK PRESSURE = 8.3 IN.H_2O GAUGE



SIMPLE-CYCLE GAS TURBINE EXHAUST GAS TEMPERATURE

GAS TEMPERATURE INTO WASTE-HEAT BOILER CORRESPONDS
TO GAS TURBINE EXHAUST TEMPERATURE

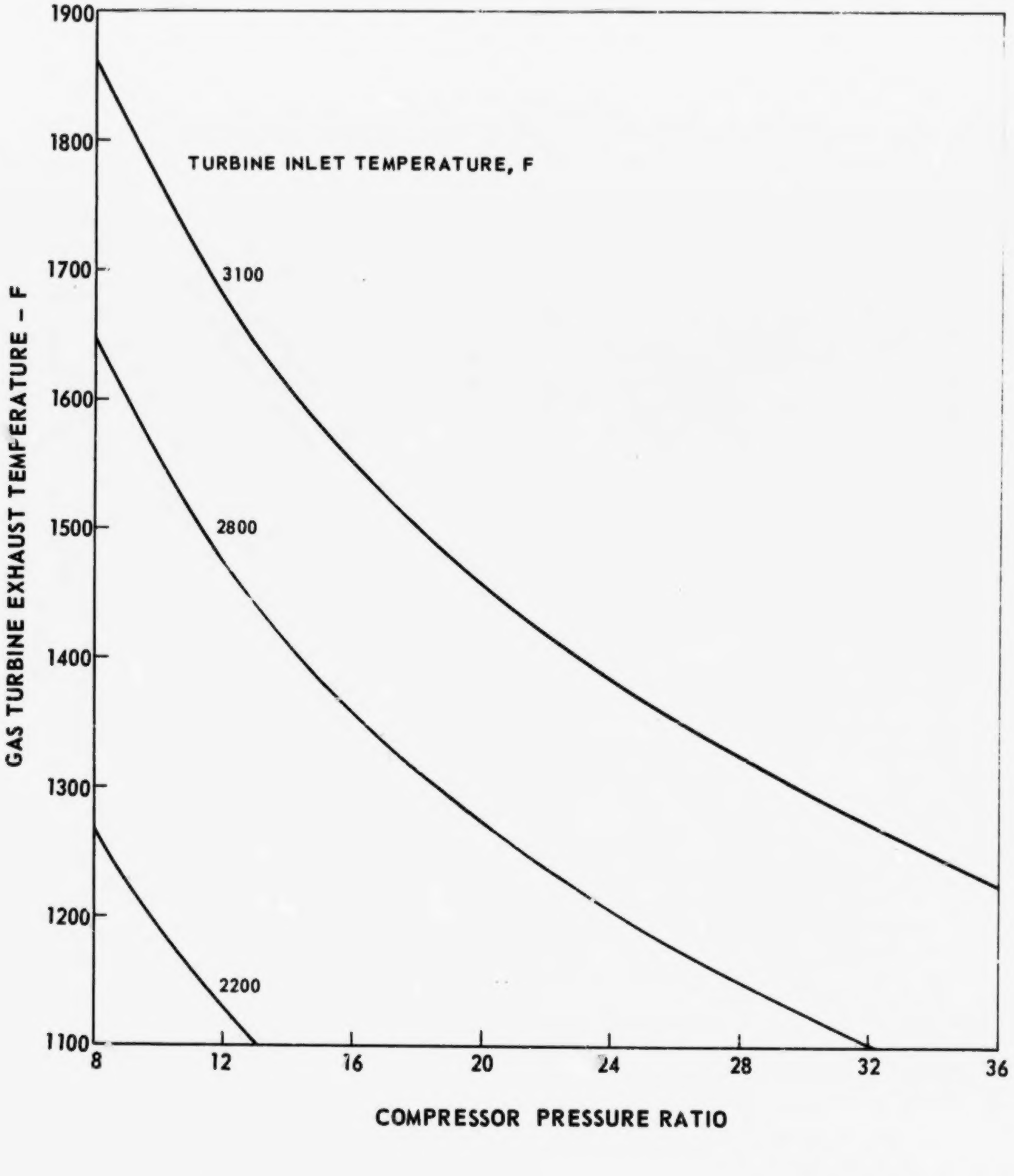
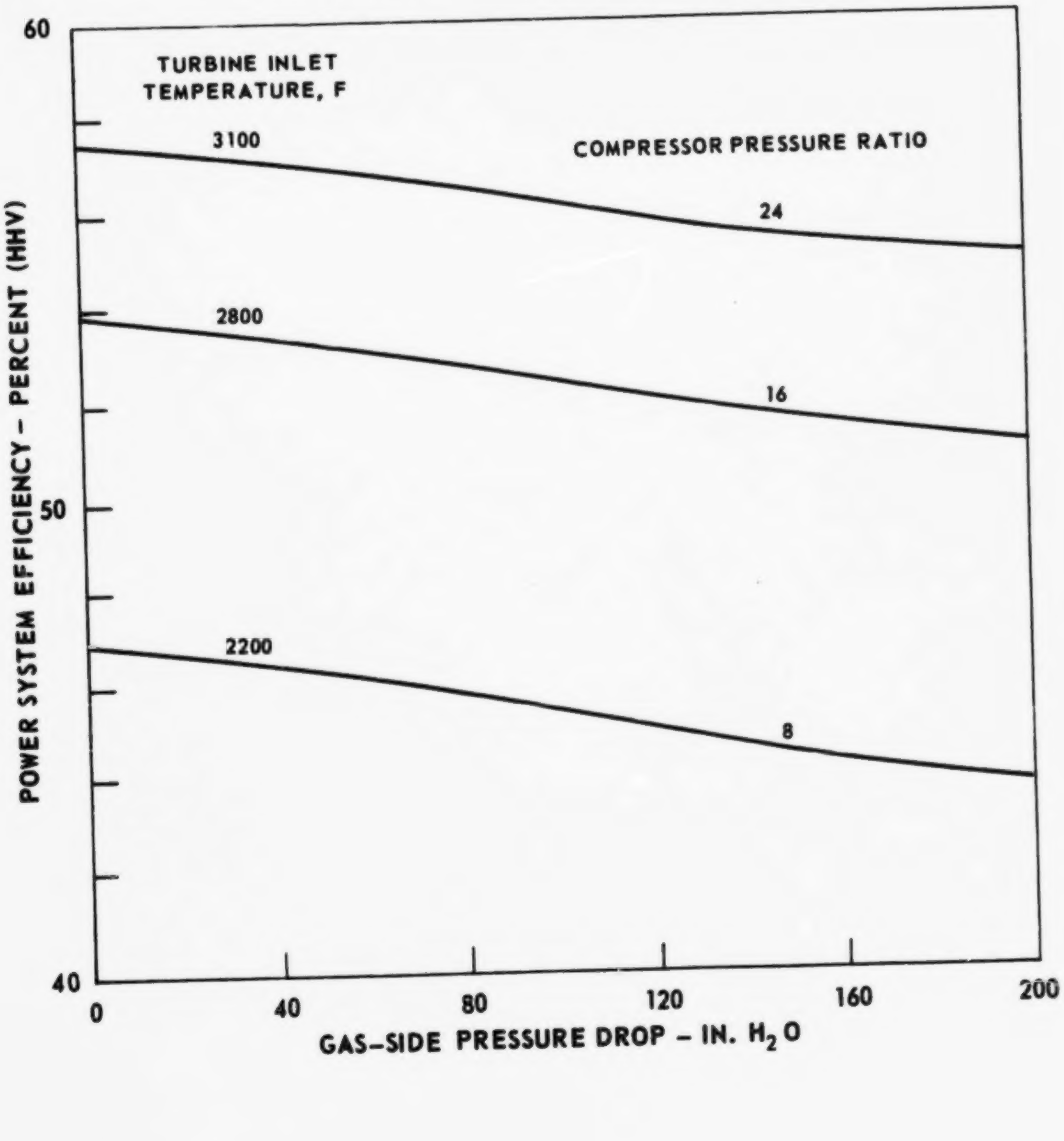


FIG. 72

EFFECT OF BOILER PRESSURE DROP ON WASTE-HEAT COGAS SYSTEM PERFORMANCE

FUEL GAS I SUPPLIED AT BURNER PRESSURE AND 150 F
STEAM CYCLE – 2400 PSIG/1000F/1000 F
STEAM CYCLE EFFICIENCY = 38.8%
GAS TURBINE COOLING AIR PRECOOLED TO 125 F



EFFECT OF FUEL SUPPLY TEMPERATURE ON WASTE-HEAT COGAS SYSTEM FUEL CONSUMPTION

TURBINE INLET TEMPERATURE = 3100 F
COMPRESSOR PRESSURE RATIO = 20
GAS TURBINE COOLING AIR PRECOOLED TO 200 F
FUEL GAS II SUPPLIED AT BURNER PRESSURE
STEAM CYCLE – 2400 PSIG/1000F/1000F

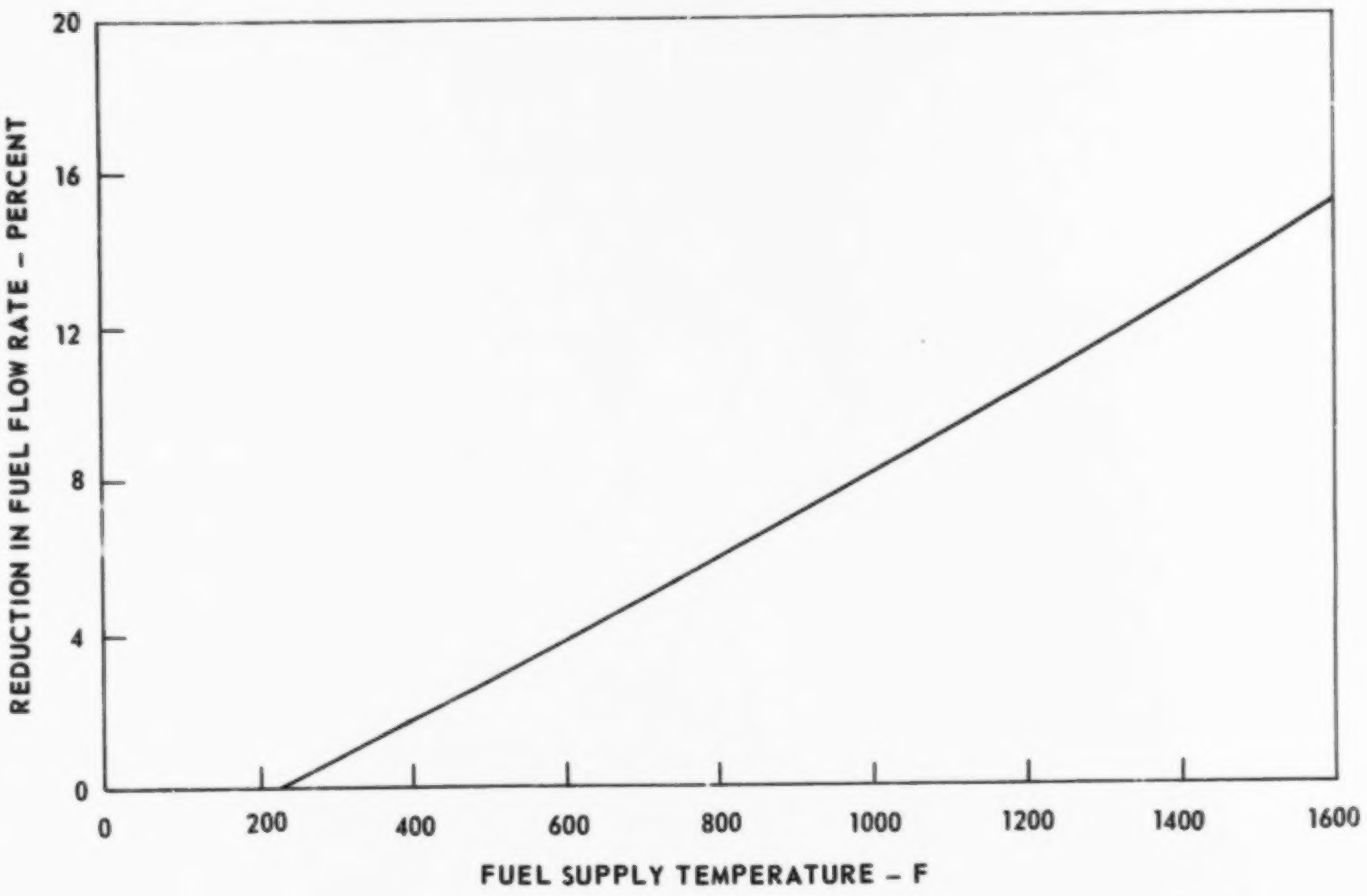


FIG. 73

FIG. 74

PART-LOAD PERFORMANCE FOR WASTE-HEAT COGAS SYSTEMS

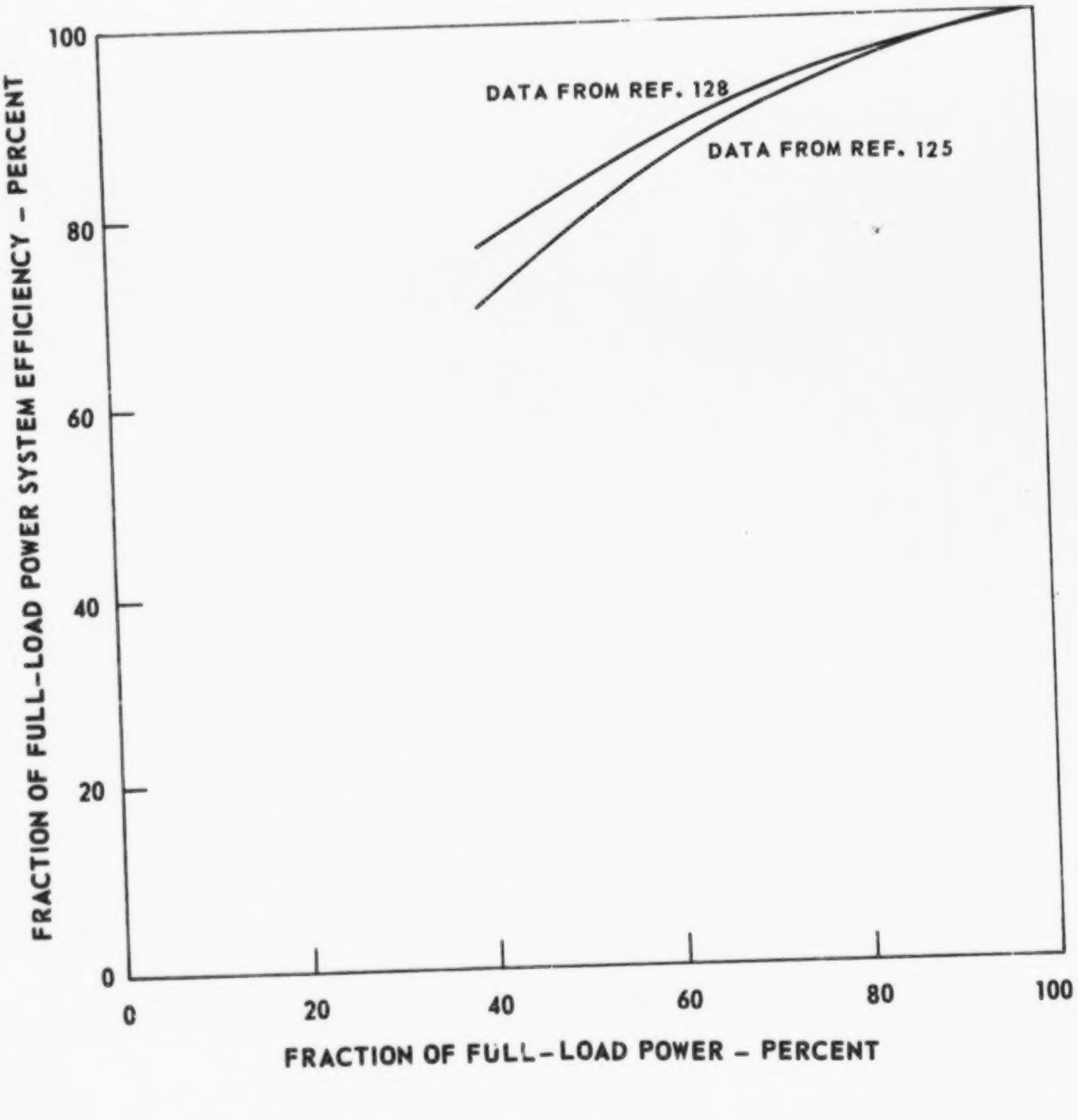


FIG. 75

WASTE-HEAT COGAS SYSTEMS WITH COMPOUND-CYCLE GAS TURBINES

TURBINE INLET TEMPERATURE = 2800 F

FUEL GAS I GAS SUPPLIED AT BURNER PRESSURE AND 150 F

STEAM CYCLE – 2400 PSIG/1000 F/1000 F

GAS TURBINE COOLING AIR PRECOOLED TO 250 F

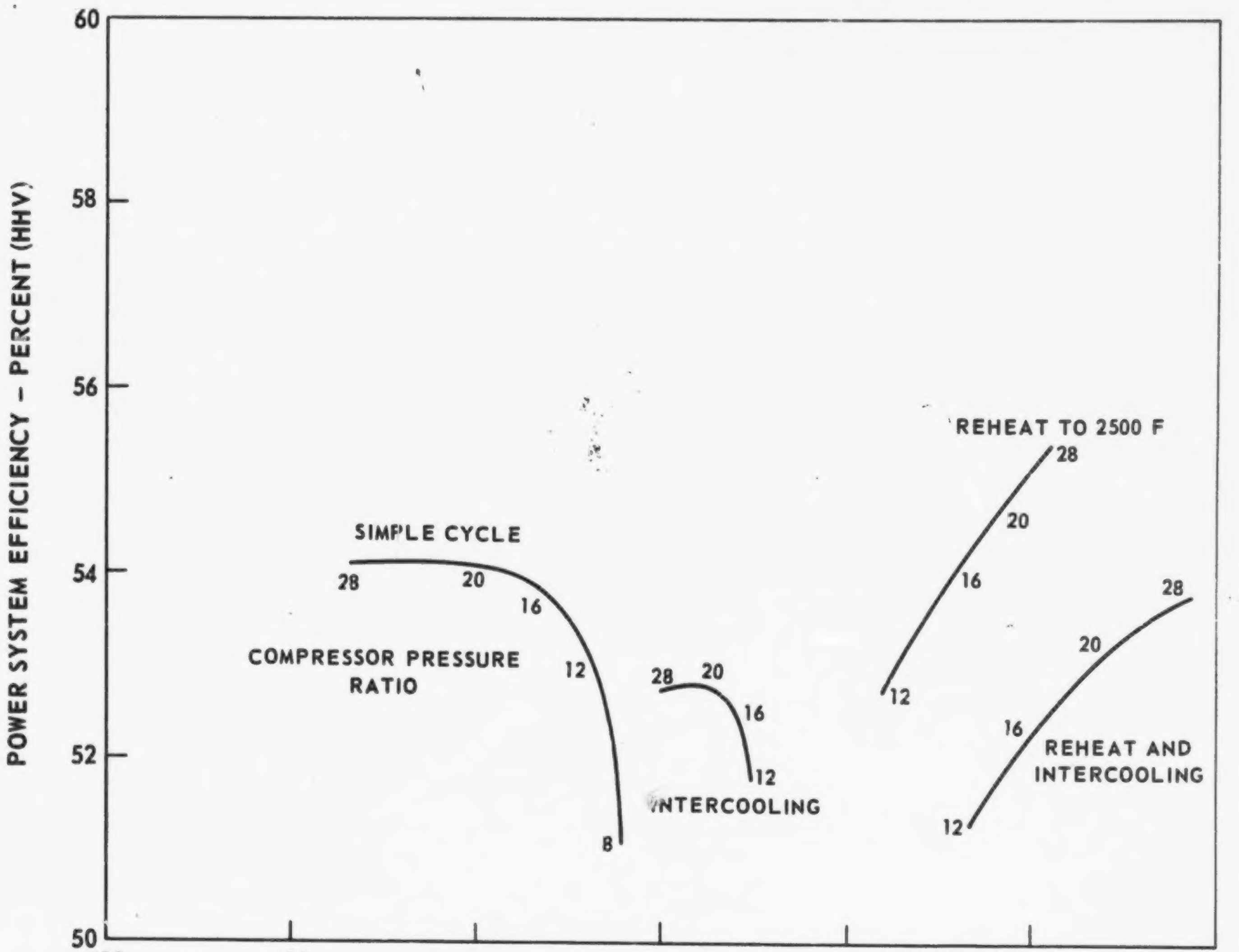
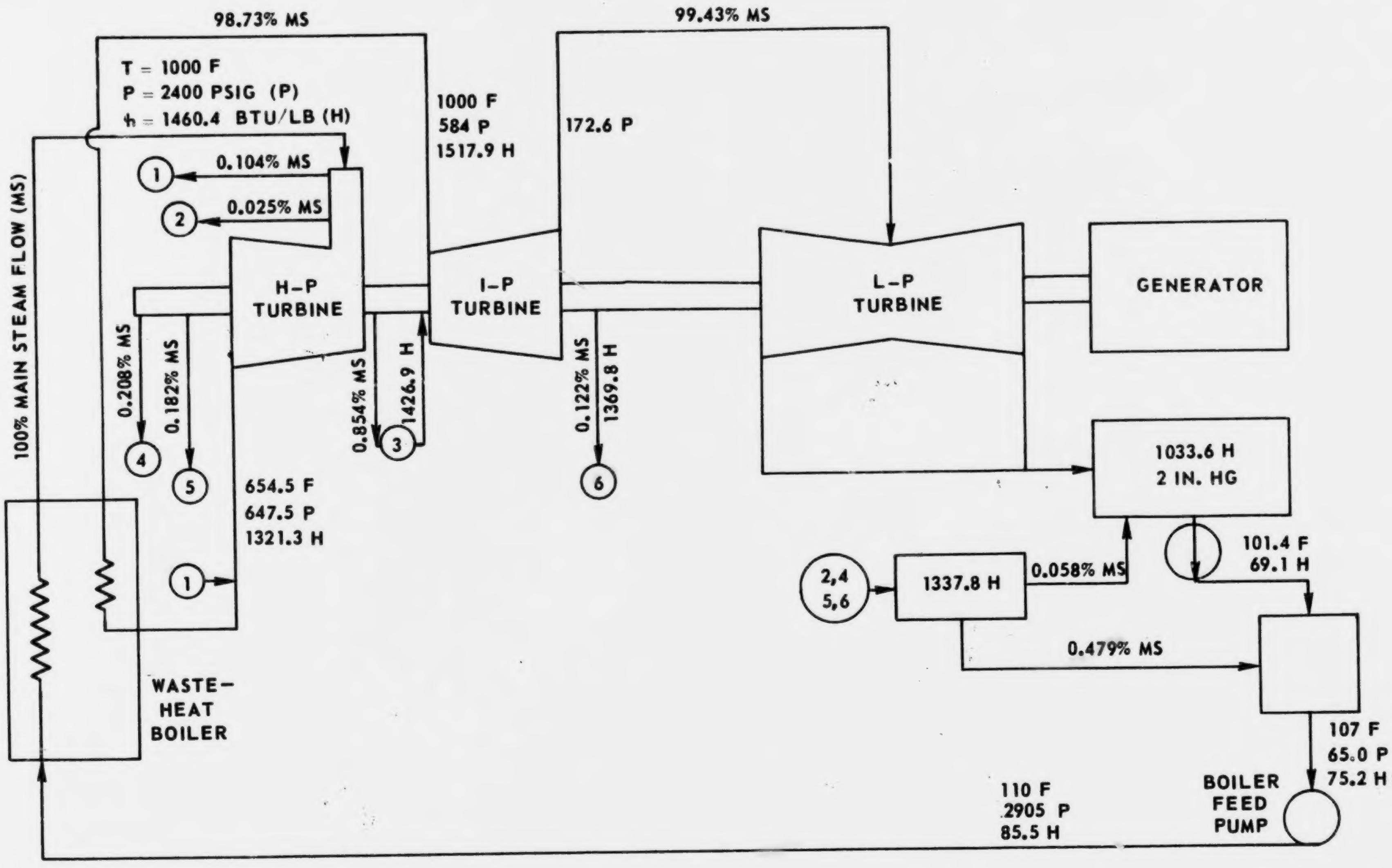


FIG. 76

HEAT BALANCE FOR 2400 PSIG/1000 F/1000 F STEAM CYCLE



STEAM TURBINE OUTPUT = 18.182 KW/100 LB MAIN STEAM FLOW
BOILER FEED PUMP POWER = 0.326 KW/100 LB MAIN STEAM FLOW
NET STEAM CYCLE OUTPUT = 17.856 KW/100 LB MAIN STEAM FLOW
NET STEAM CYCLE HEAT RATE = 8787 BTU/KW-HR
NET STEAM CYCLE EFFICIENCY = 38.83%
NET THROTTLE STEAM RATE = 5.60 LB/KW-HR

FIG. 77

WASTE-HEAT BOILER STACK TEMPERATURE

STEAM CYCLE – 2400 PSIG/1000 F/1000 F
FEEDWATER TEMPERATURE = 110 F
STEAM CYCLE EFFICIENCY = 38.8%
PINCH TEMPERATURE DIFFERENCE = 50 F

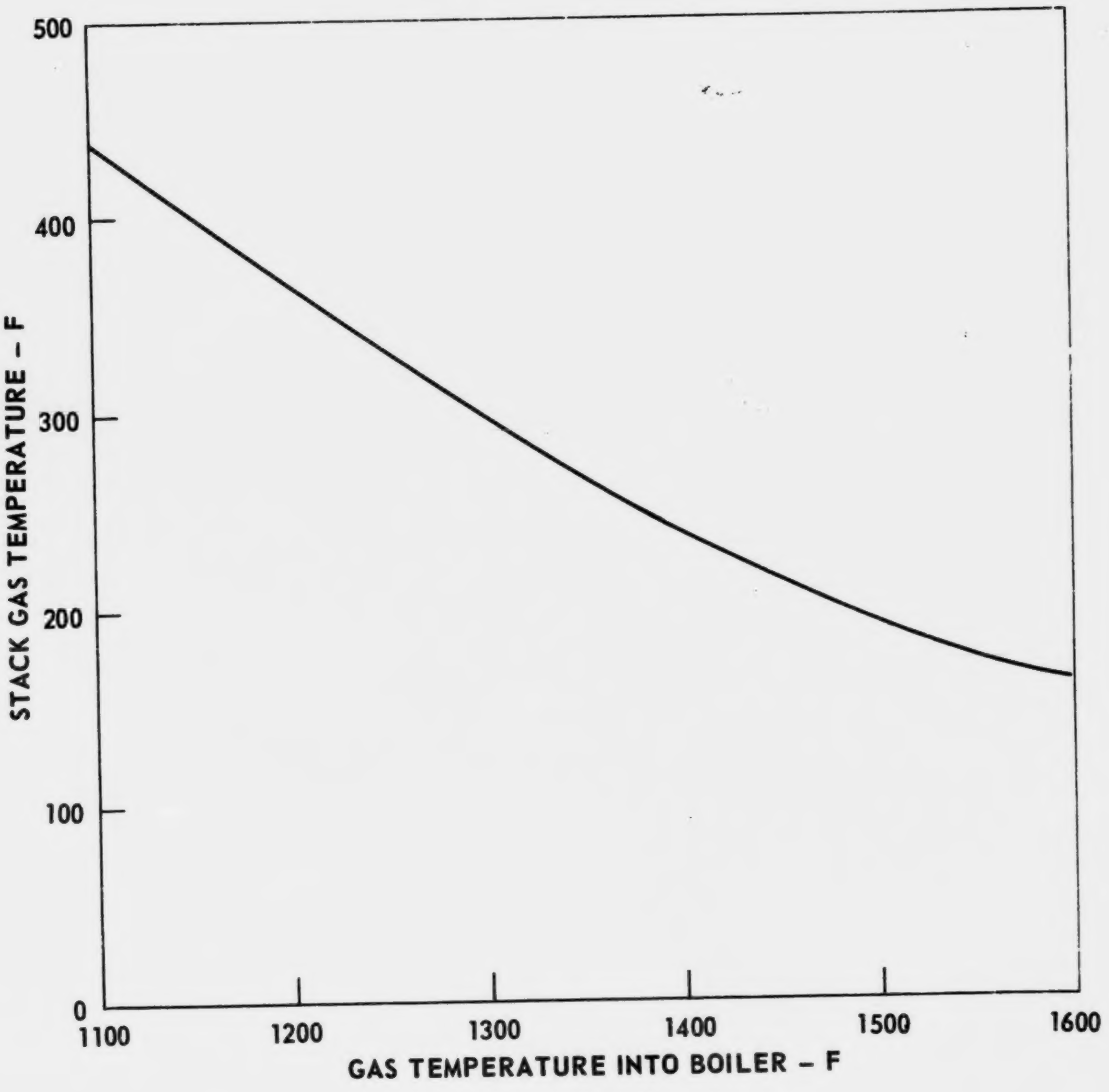


FIG. 78

WASTE-HEAT BOILER STEAM GENERATION RATE

STEAM CYCLE – 2400 PSIG/1000 F/1000 F

PINCH TEMPERATURE DIFFERENCE = 50 F

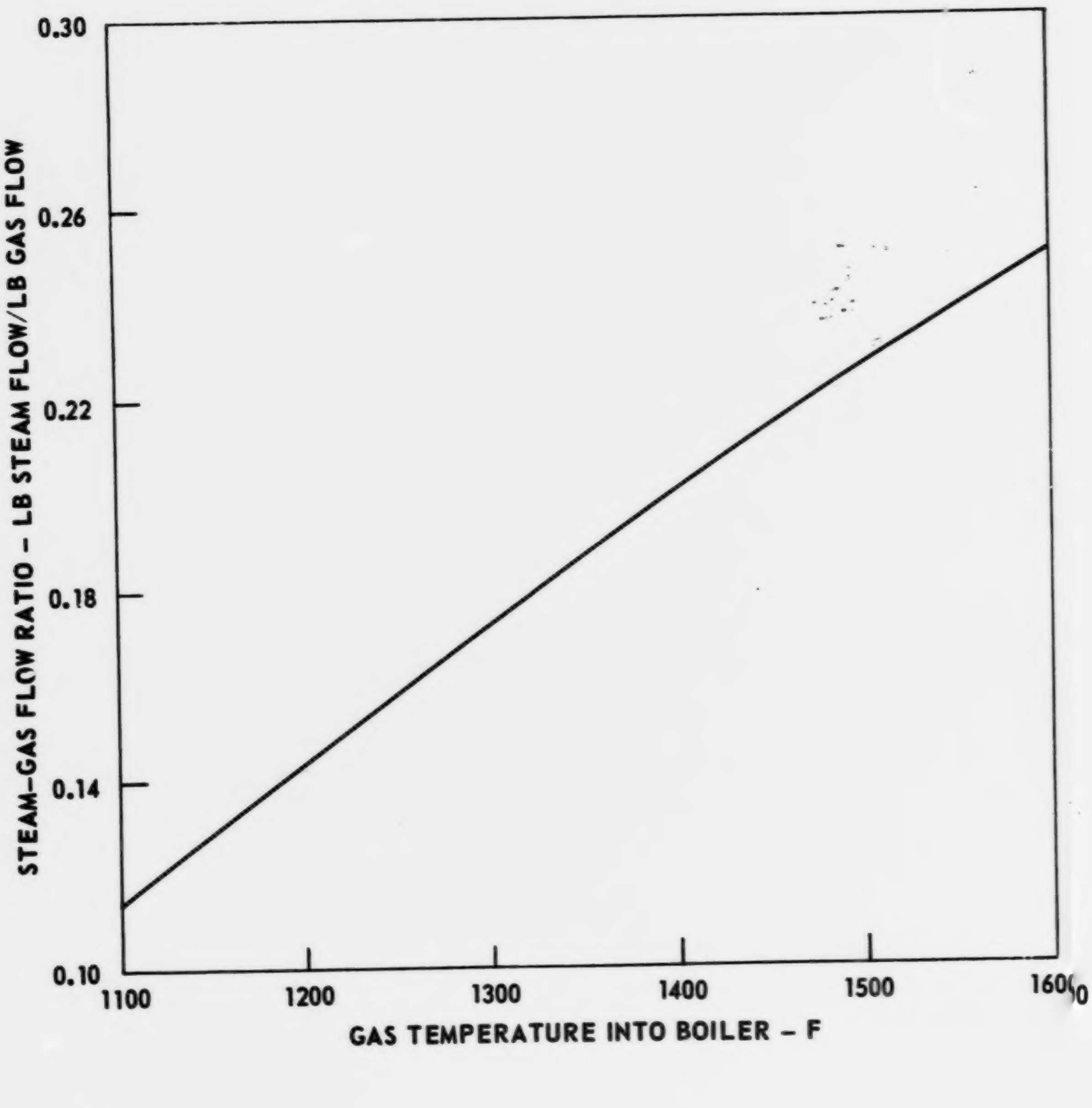


FIG. 79

PERFORMANCE OF WASTE-HEAT STEAM SYSTEM

STEAM CYCLE – 2400 PSIG/1000 F/1000 F
FEEDWATER TEMPERATURE = 110 F
STEAM CYCLE EFFICIENCY = 38.8%
PINCH TEMPERATURE DIFFERENCE = 50 F

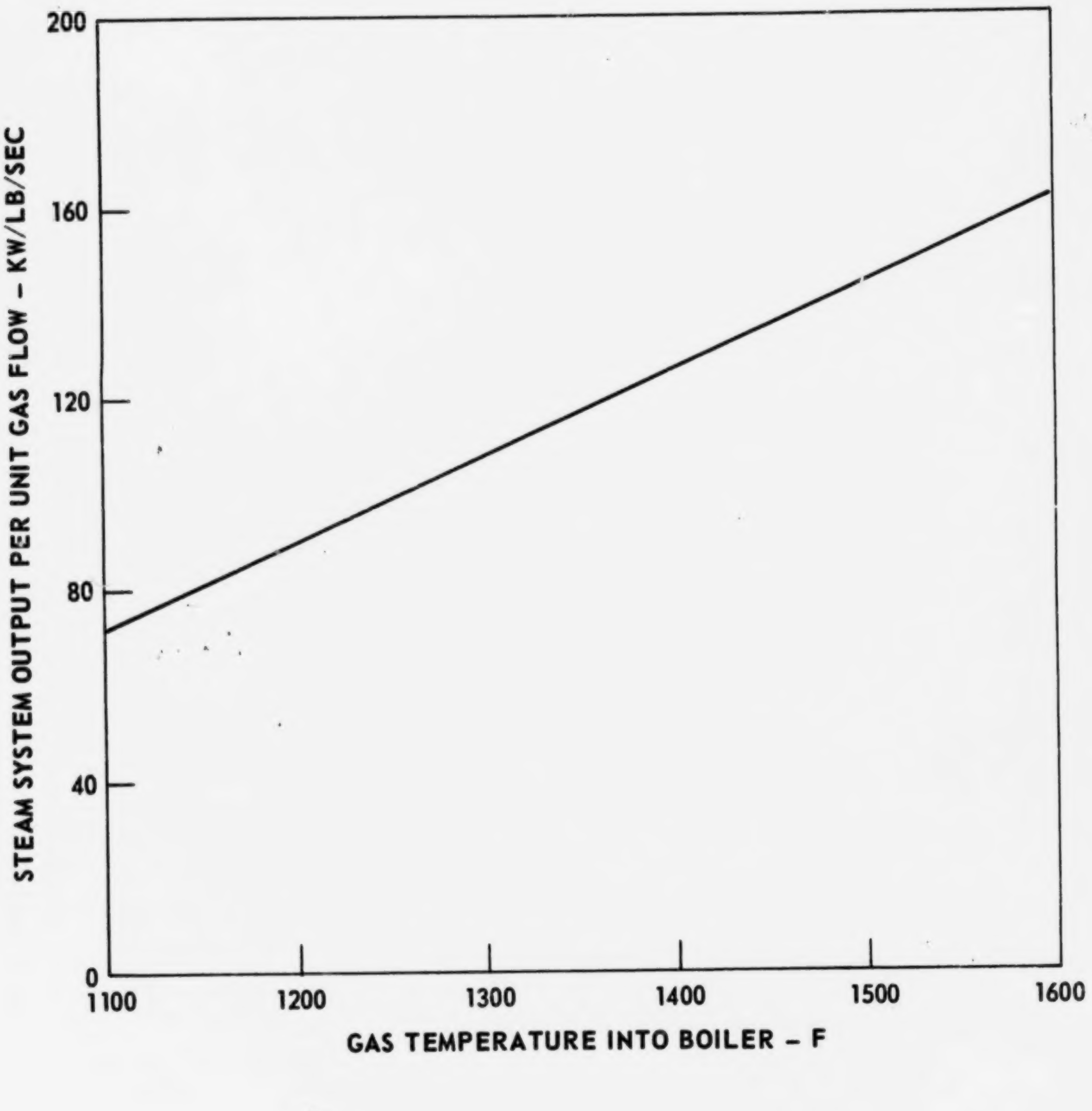
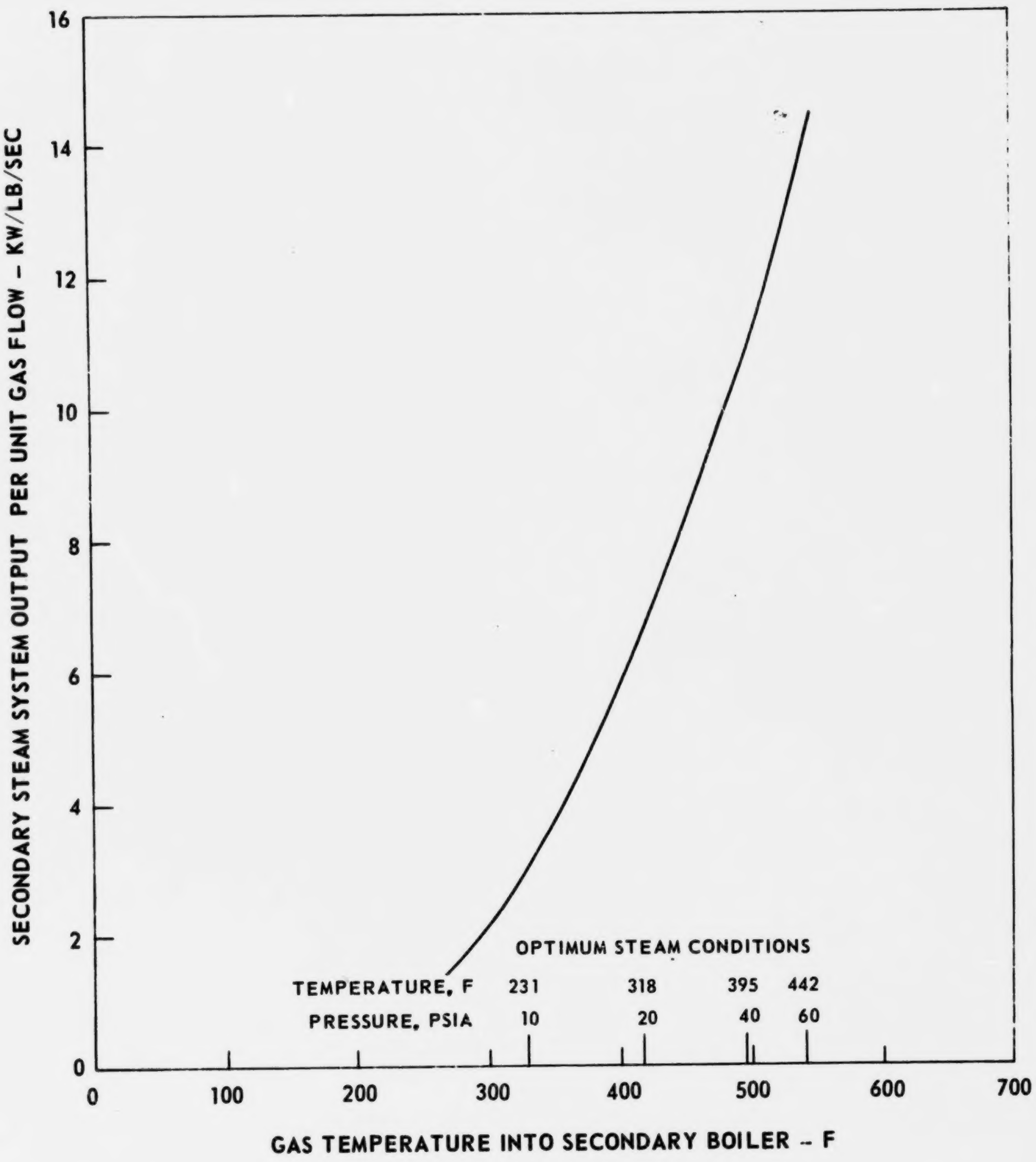


FIG. 80

PERFORMANCE OF SECONDARY STEAM SYSTEMS

GAS SPECIFIC HEAT = 0.27 BTU/LB F



TOTAL WASTE-HEAT BOILER HEAT TRANSFER SURFACE AREA

1000 – MW NOMINAL COGAS SYSTEM

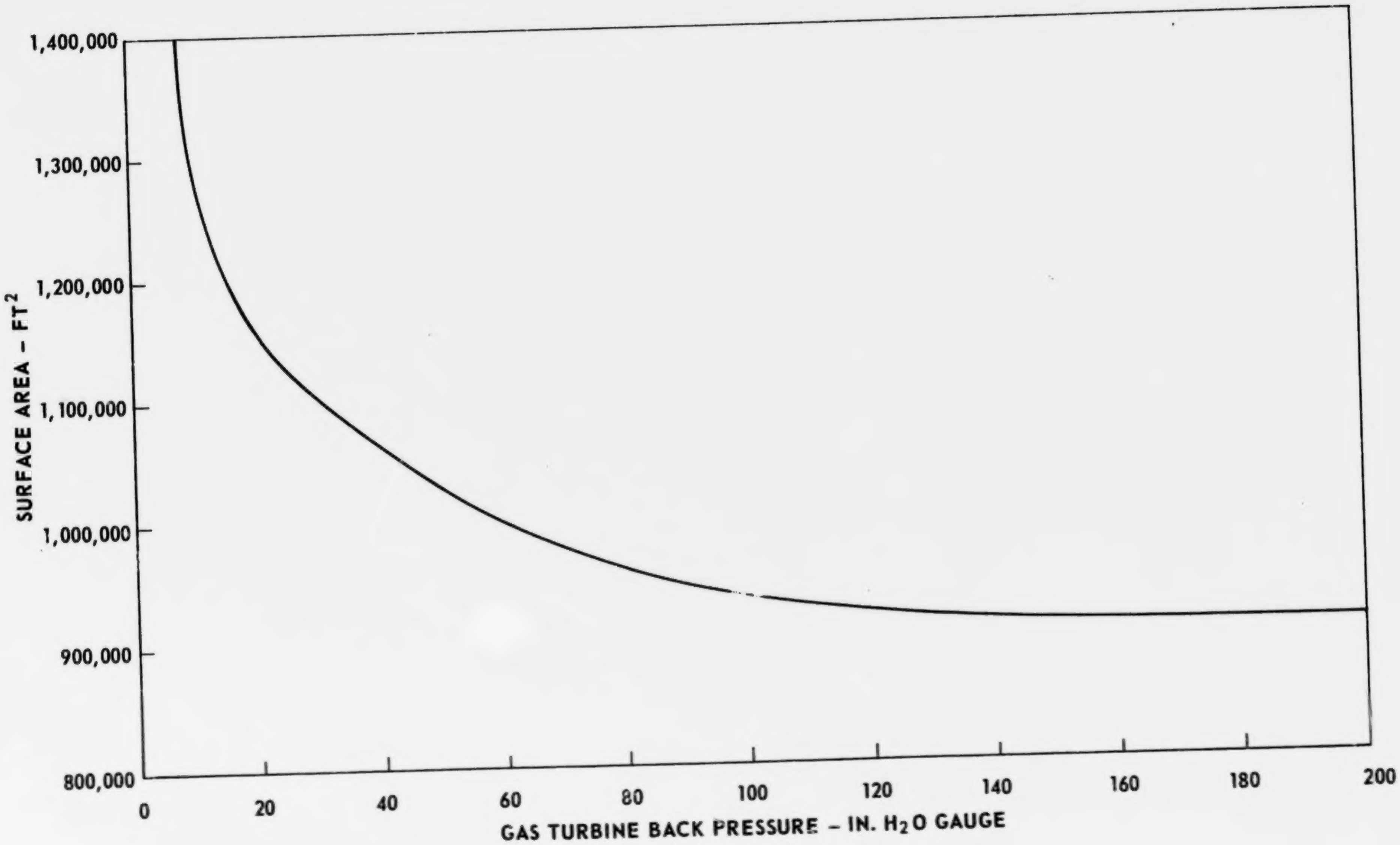
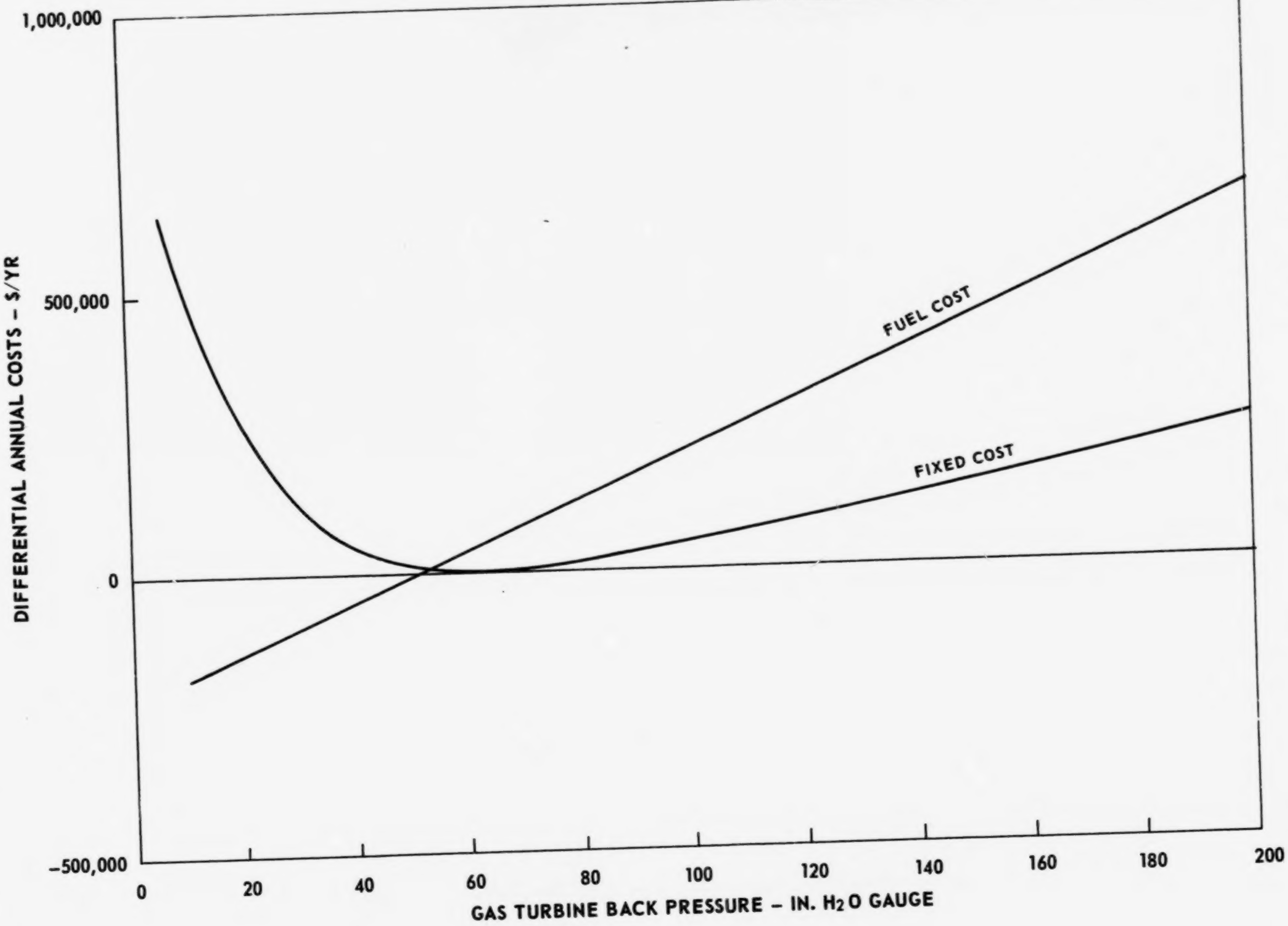


FIG. 81

FIG. 82

DIFFERENTIAL ANNUAL FUEL COSTS AND FIXED CHARGES FOR SECOND-GENERATION COGAS SYSTEMS

1000 – MW NOMINAL COGAS SYSTEM



TOTAL DIFFERENTIAL ANNUAL EVALUATED COSTS FOR SECOND-GENERATION COGAS SYSTEMS

1000 – MW NOMINAL COGAS SYSTEM

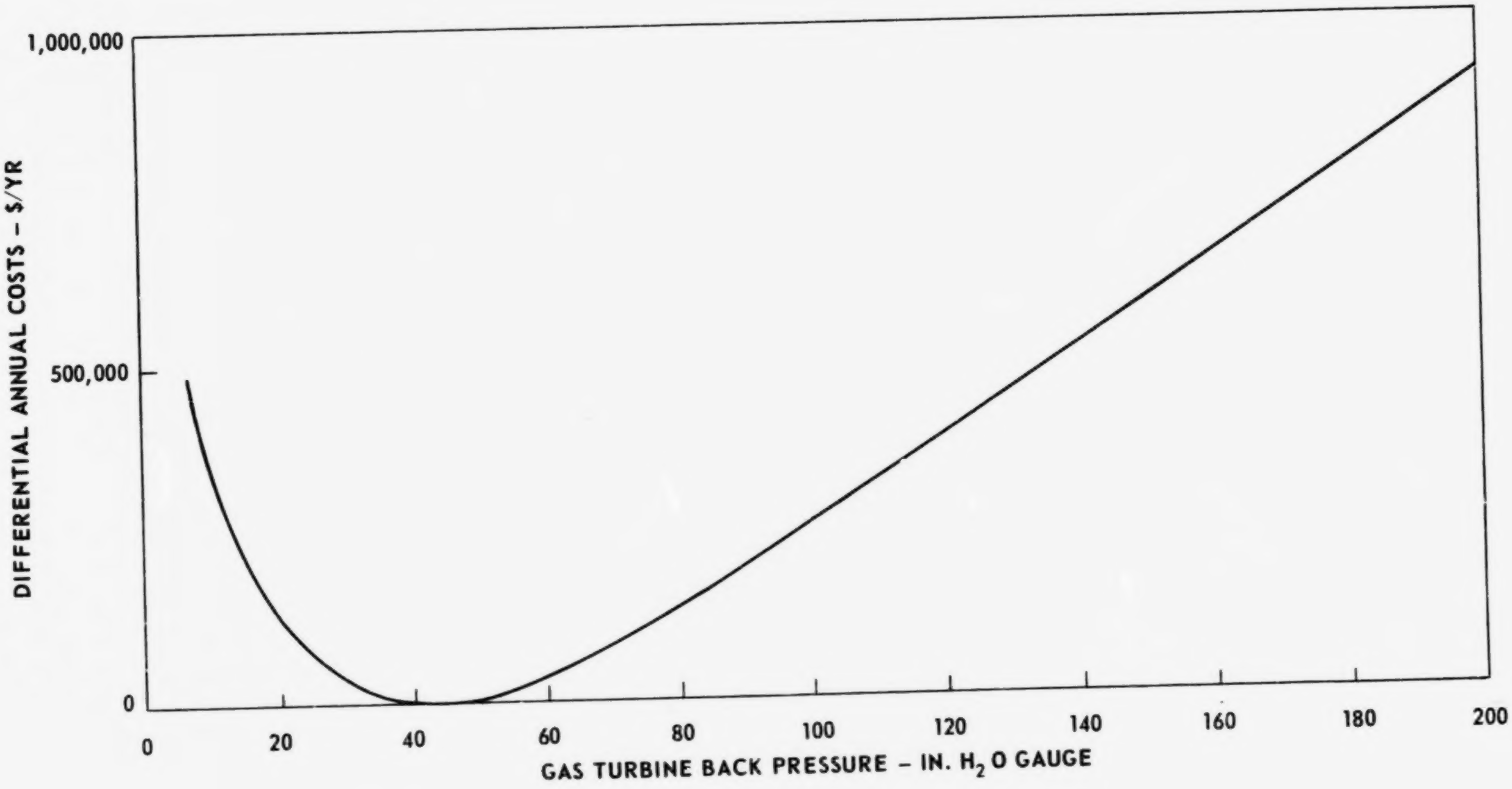
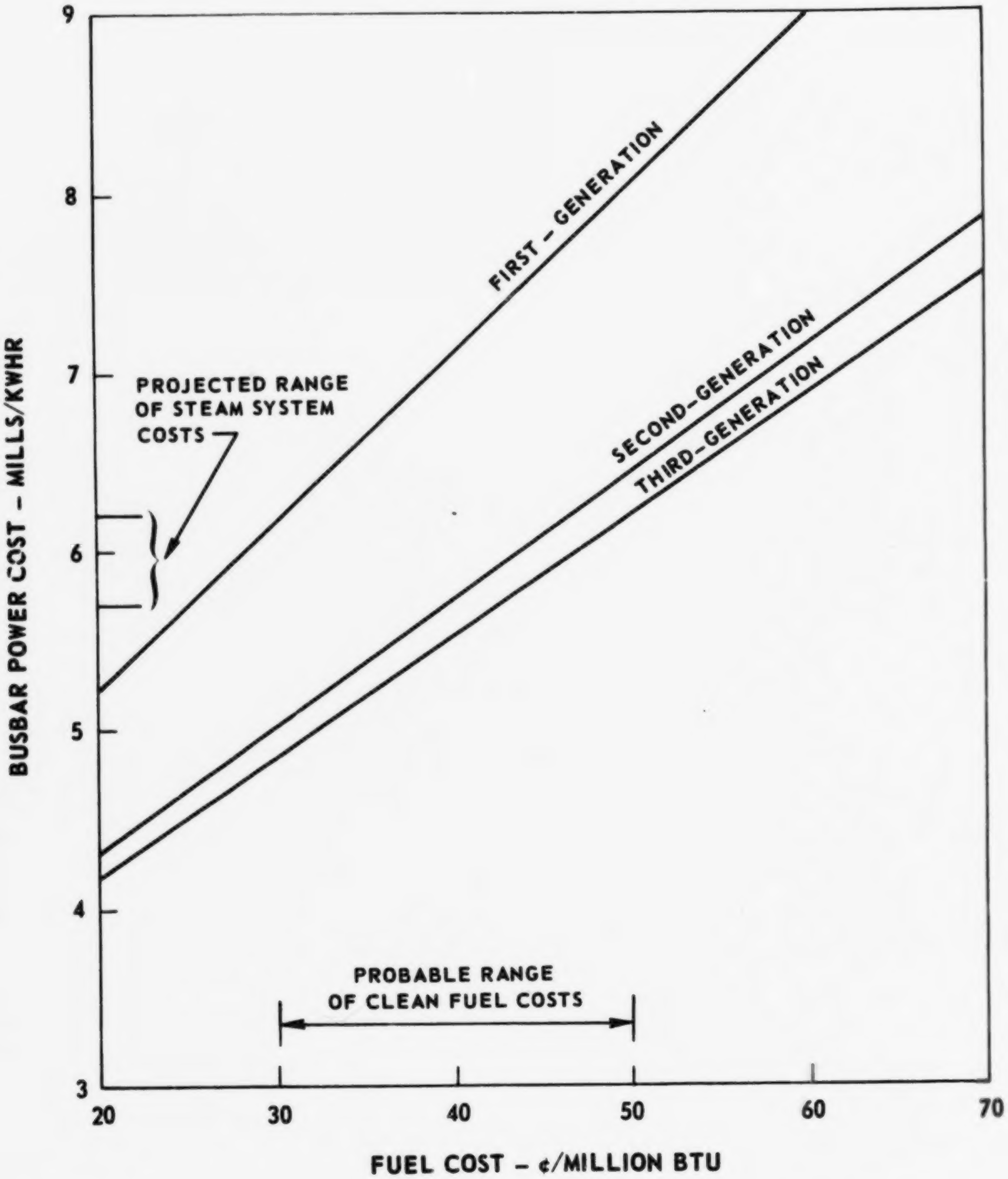


FIG. 83

FIG. 84

COMPARISON OF POWER COSTS FOR COGAS POWER SYSTEMS



TYPICAL POTASSIUM TOPPING CYCLE

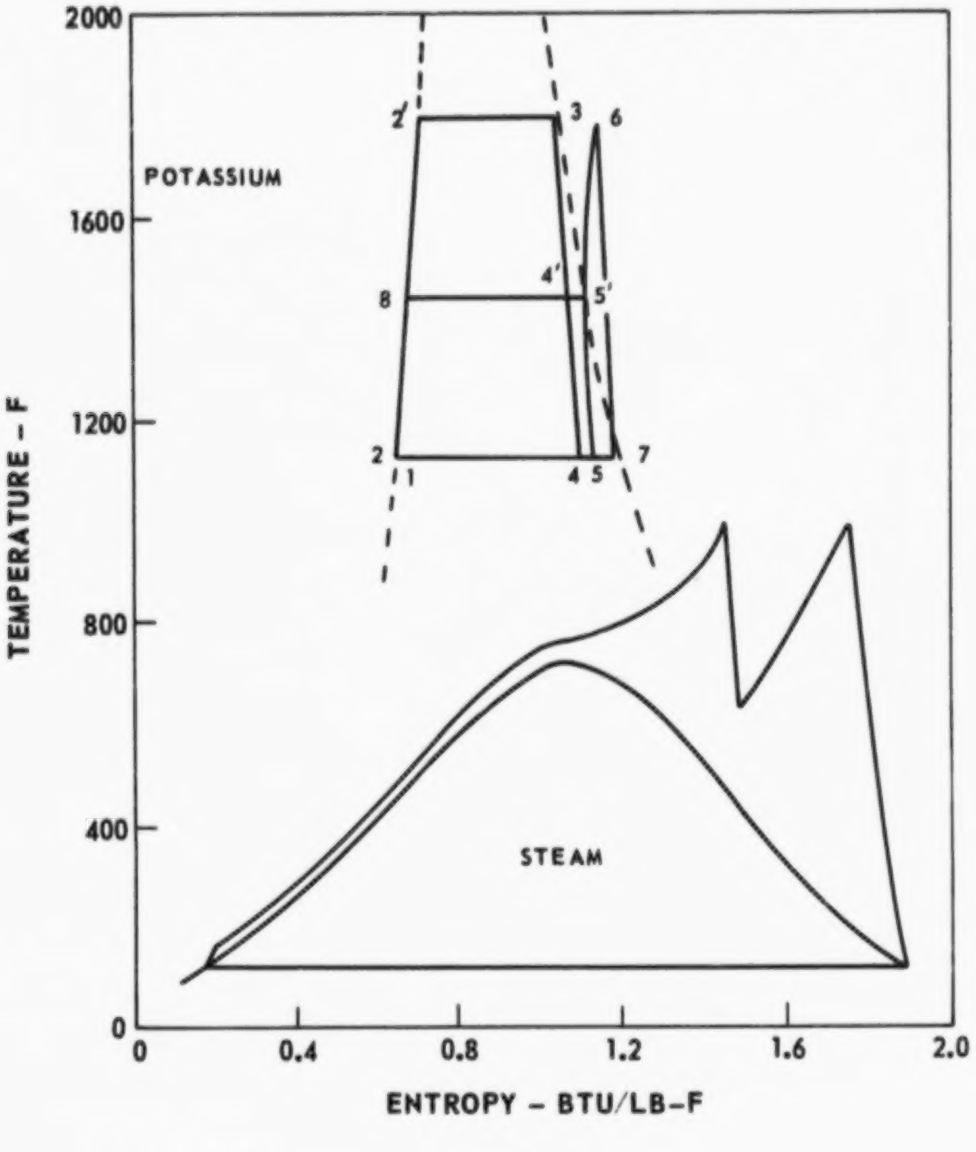


FIG. 86

BINARY CYCLE THERMAL EFFICIENCY

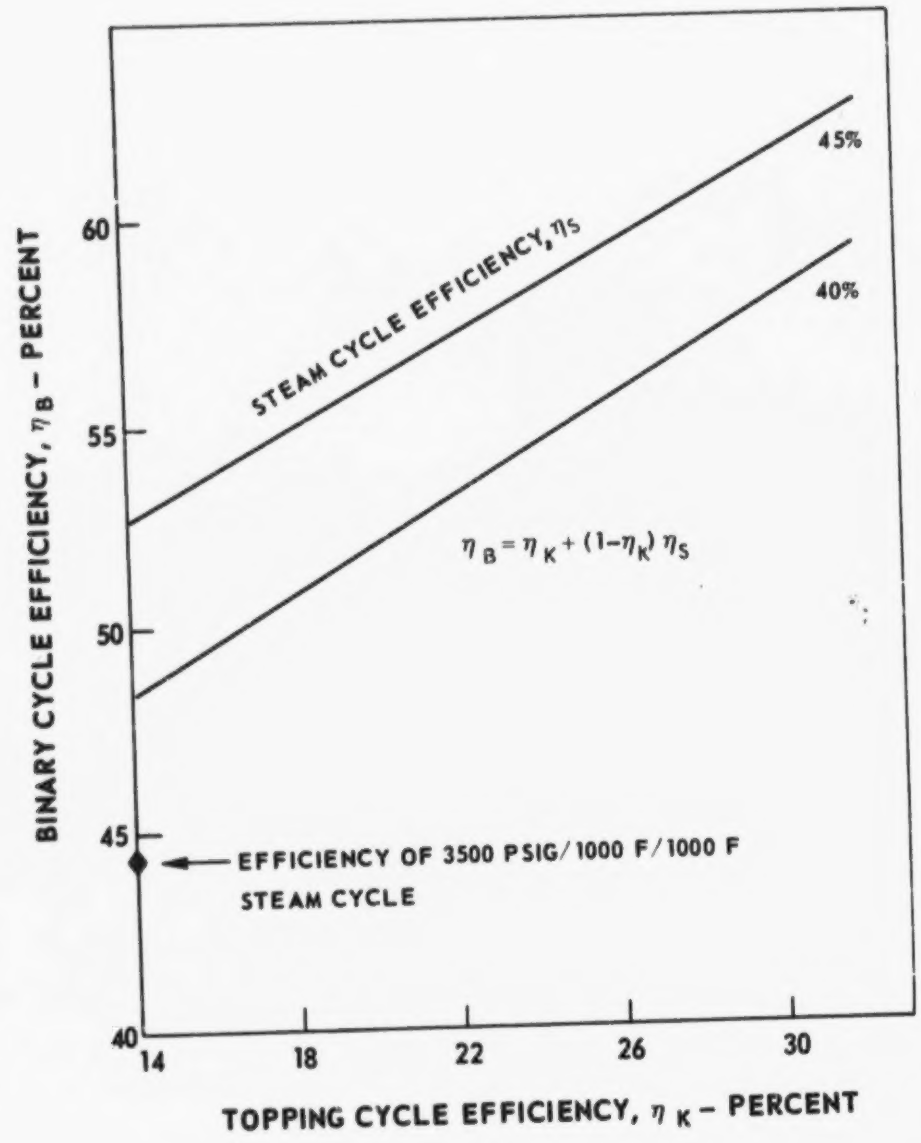


FIG. 87

LOW-TEMPERATURE POTASSIUM TOPPING CYCLE

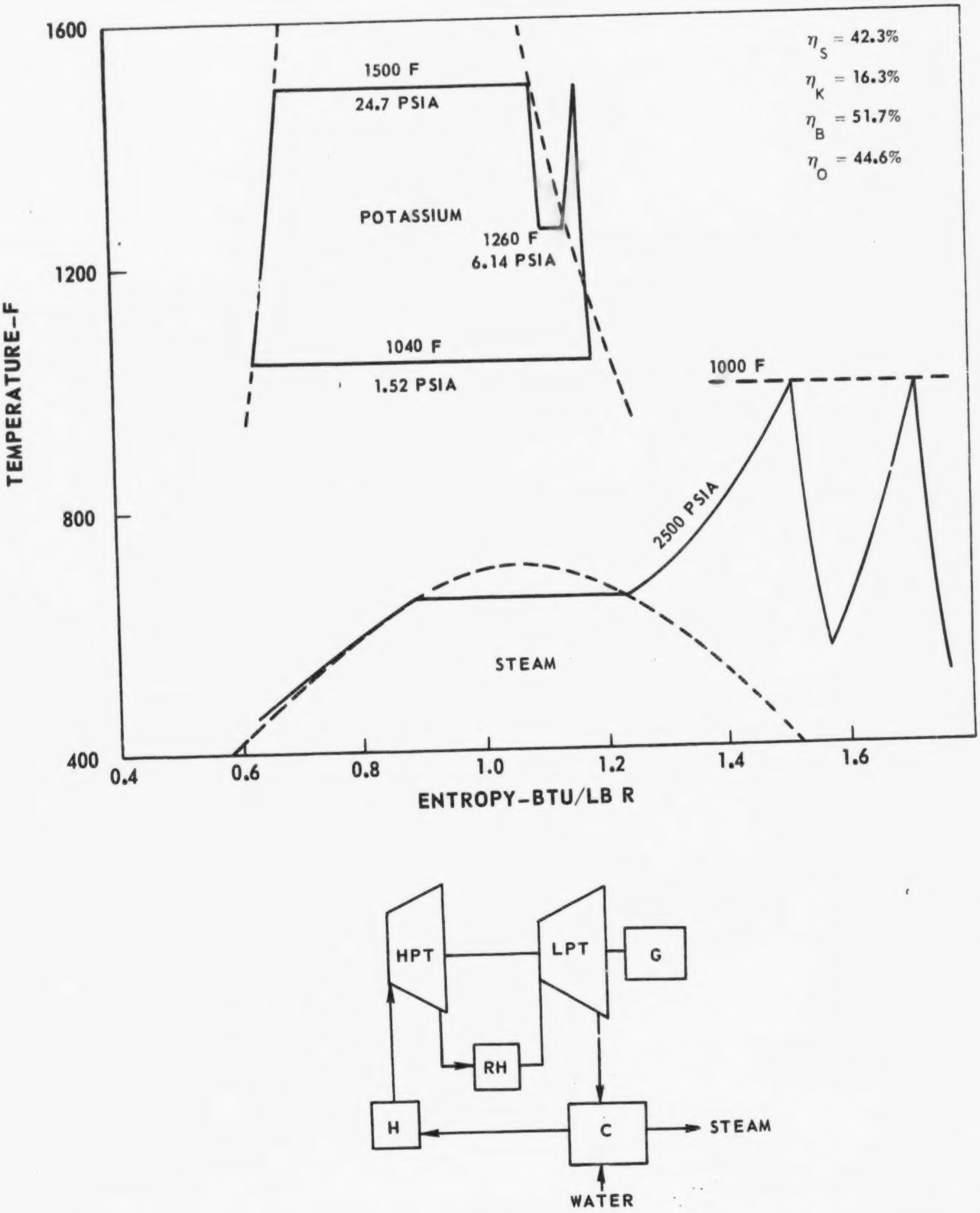


FIG. 88

PERFORMANCE OF LOW-TEMPERATURE POTASSIUM TOPPING CYCLE

POTASSIUM TURBINE EFFICIENCY – 75%

POTASSIUM PUMP EFFICIENCY – 70%

POTASSIUM REHEAT PRESSURE – $\sqrt{P_1/P_3}$

STEAM CYCLE – 2500 PSIA/1000 F/1000 F

$\eta_S = 42.3\%$

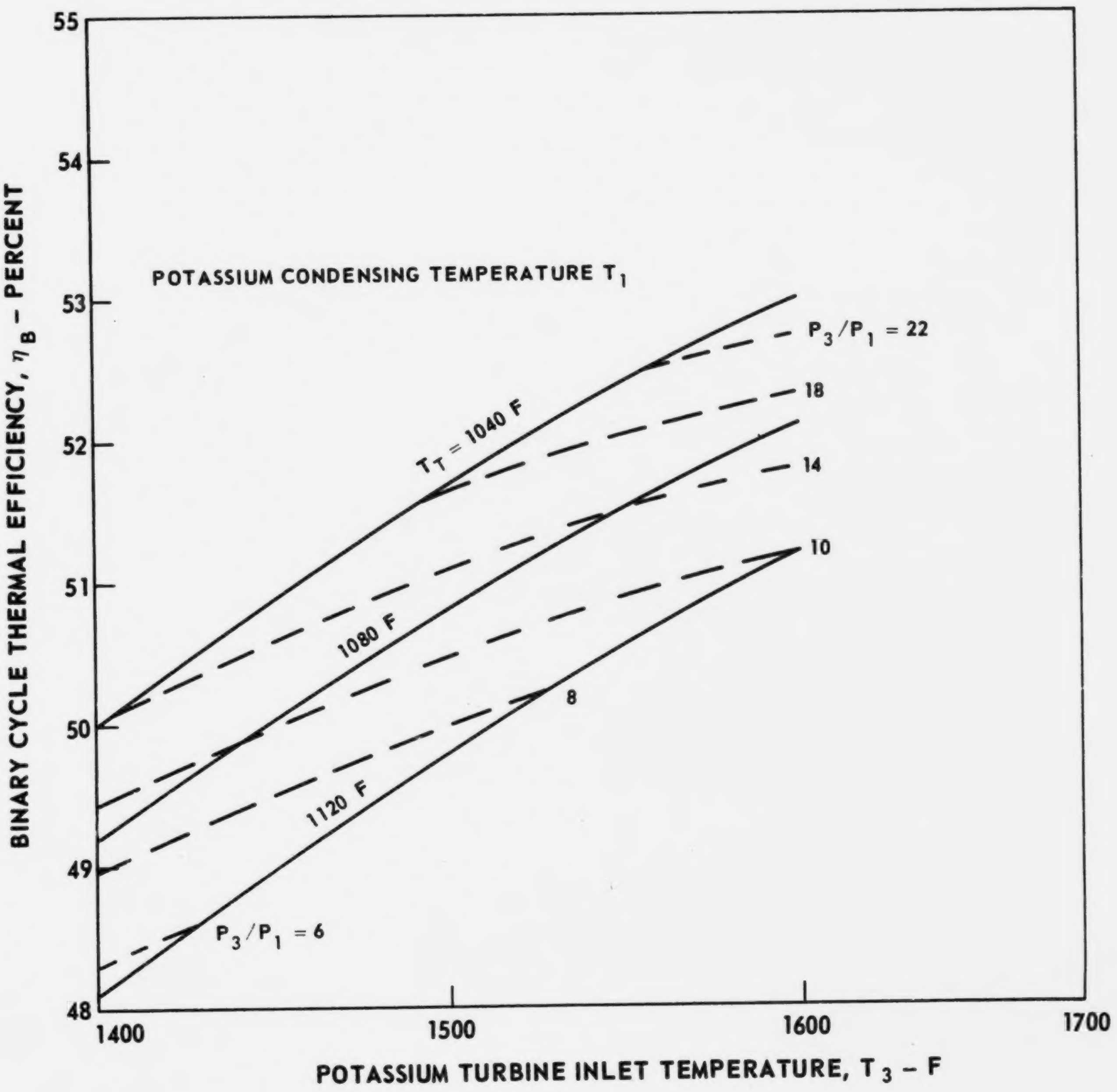


FIG. 89

HIGH-TEMPERATURE POTASSIUM TOPPING CYCLE

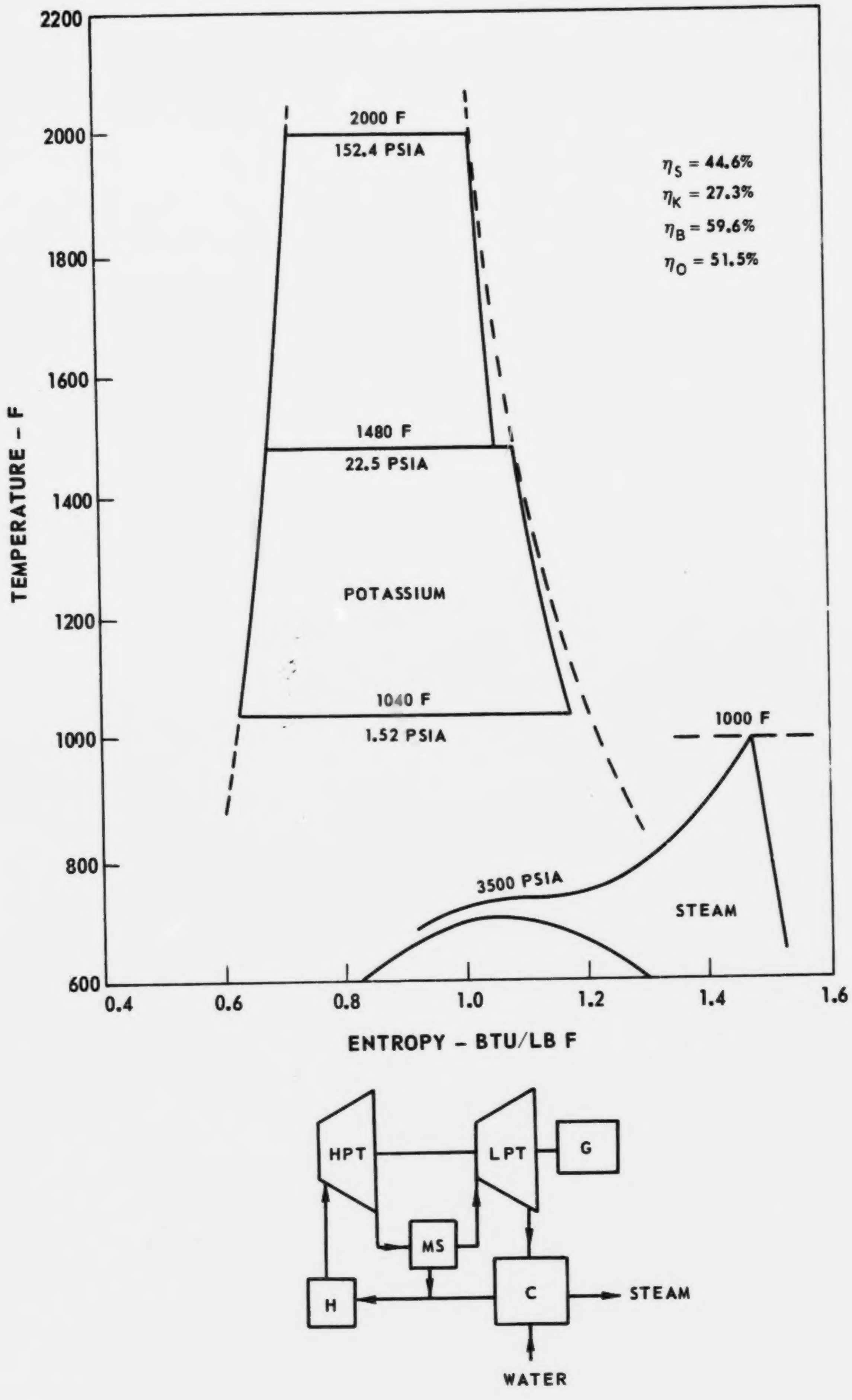


FIG. 90

COMPARISON OF POWER COSTS FOR POTASSIUM TOPPING SYSTEM

1000-MW NOMINAL SIZE

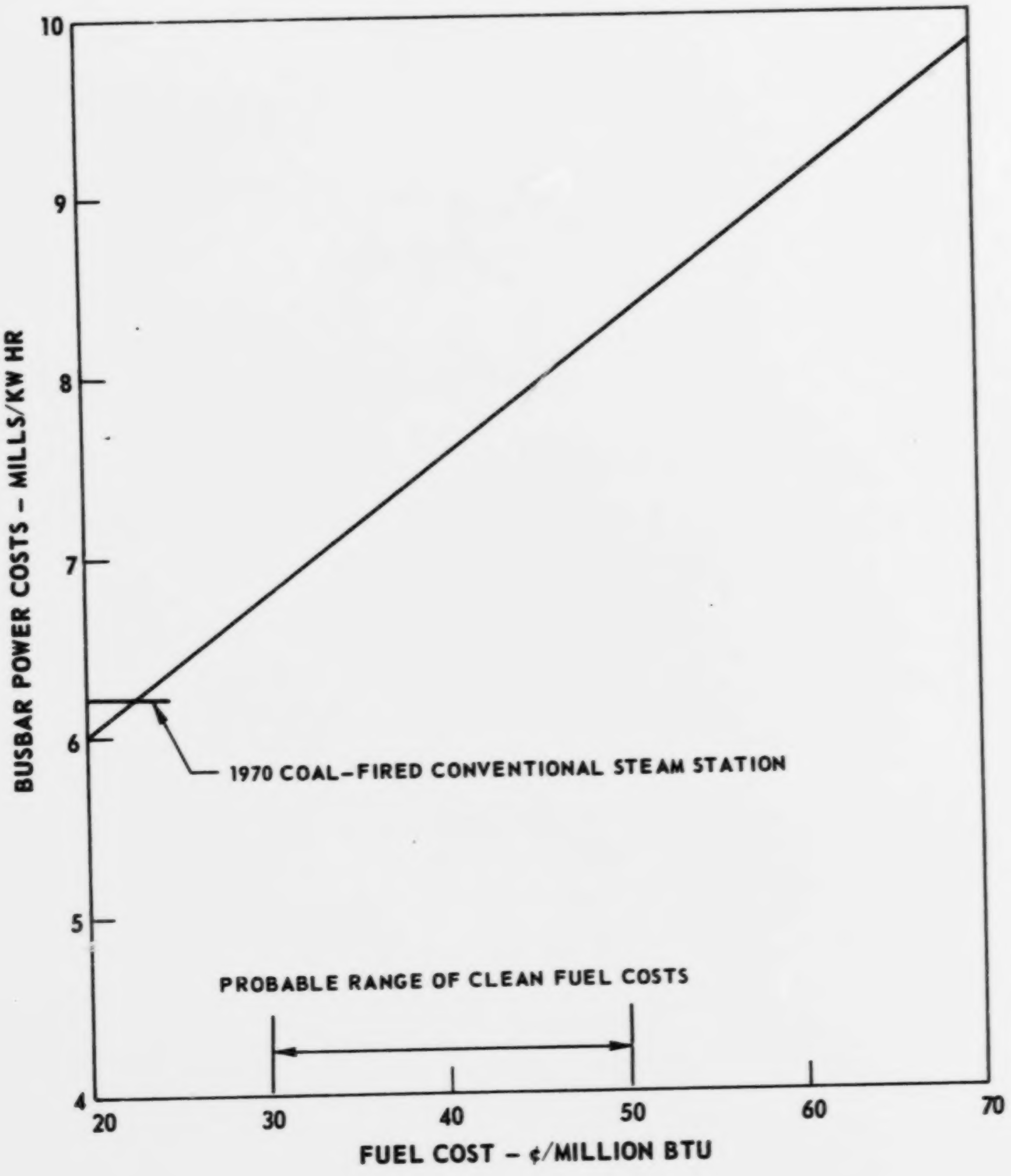


FIG. 91

BOTTOMING CYCLE APPLICATIONS

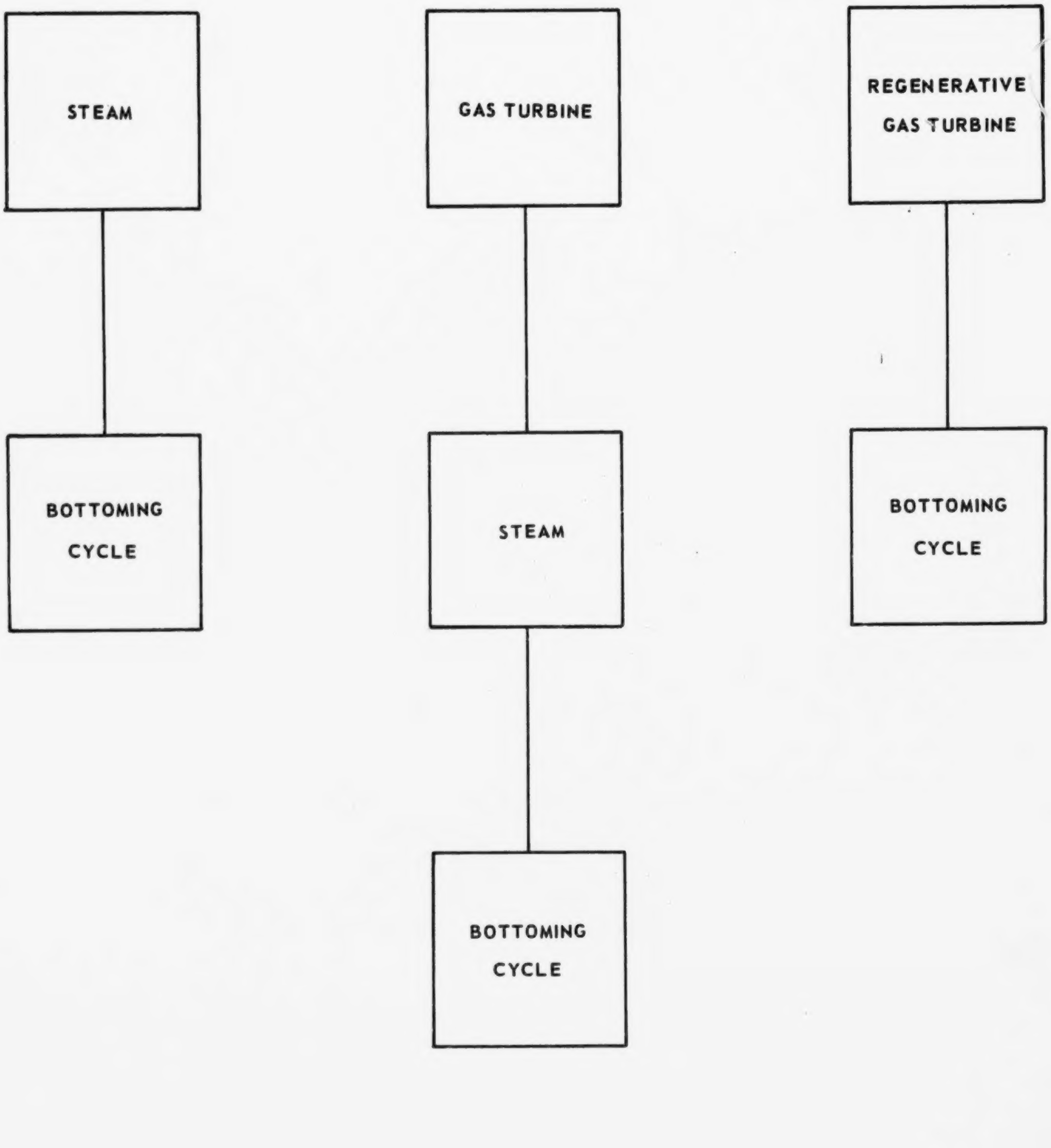


FIG. 92

AMMONIA BOTTOMING CYCLE

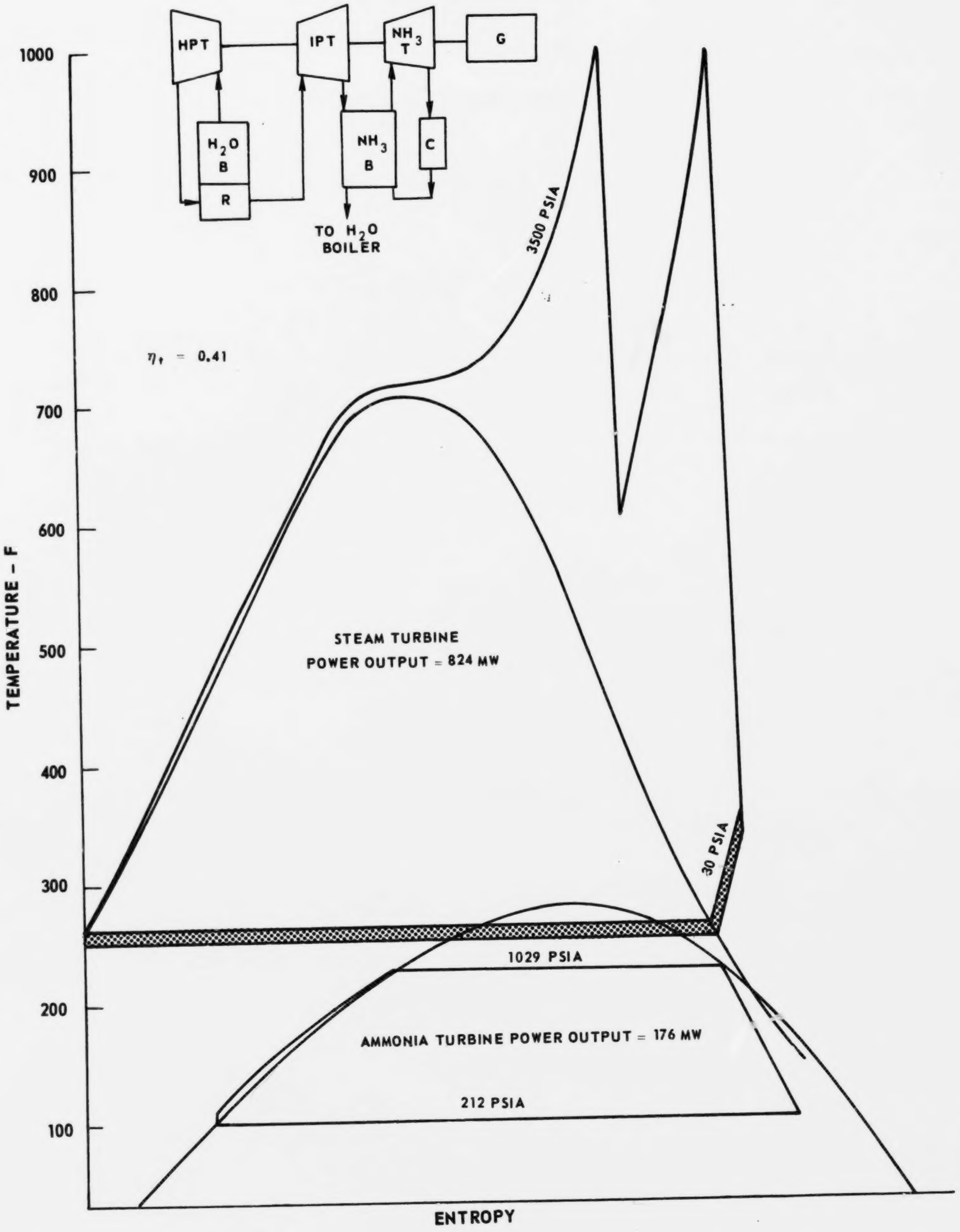


FIG. 93

BOTTOMING CYCLE PERFORMANCE WITH ORGANIC WORKING FLUIDS

PRESSURE LOSS = 5%
BOILER PRESSURE LOSS = 10%
SIMPLE CYCLE – NO FEED HEATING, ETC.
ASSUMED TURBINE ISENTROPIC EFFICIENCY, η_T = 80%
CONDENSER TEMPERATURE = 120 F

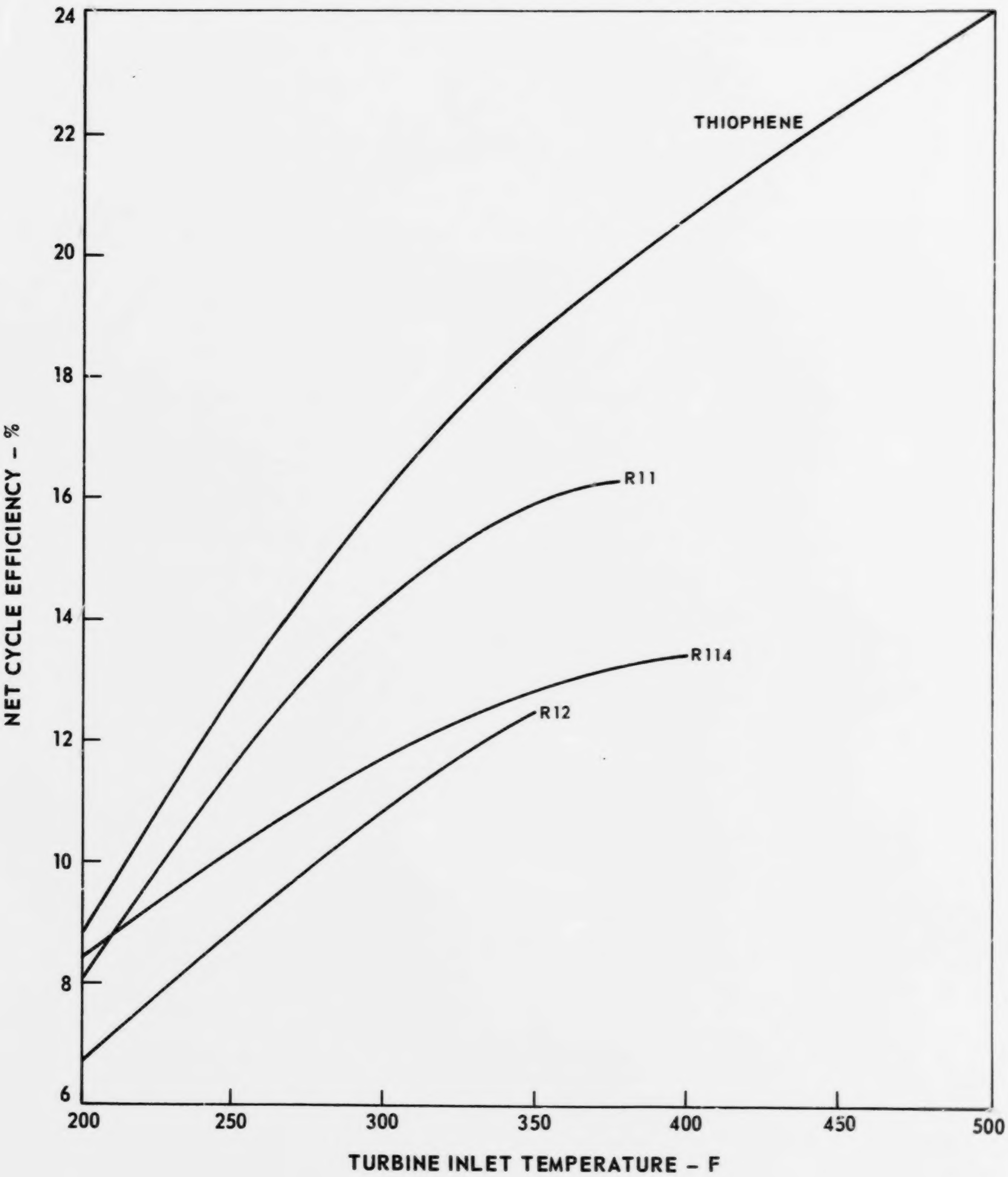


FIG. 94

COMPARISON OF STEAM AND R11 BOTTOMING CYCLES

SENSIBLE HEAT CAPACITY OF WASTE HEAT SOURCE, $w\,c_P = \frac{Q}{\Delta T} = 200{,}000$ Btu/hr F

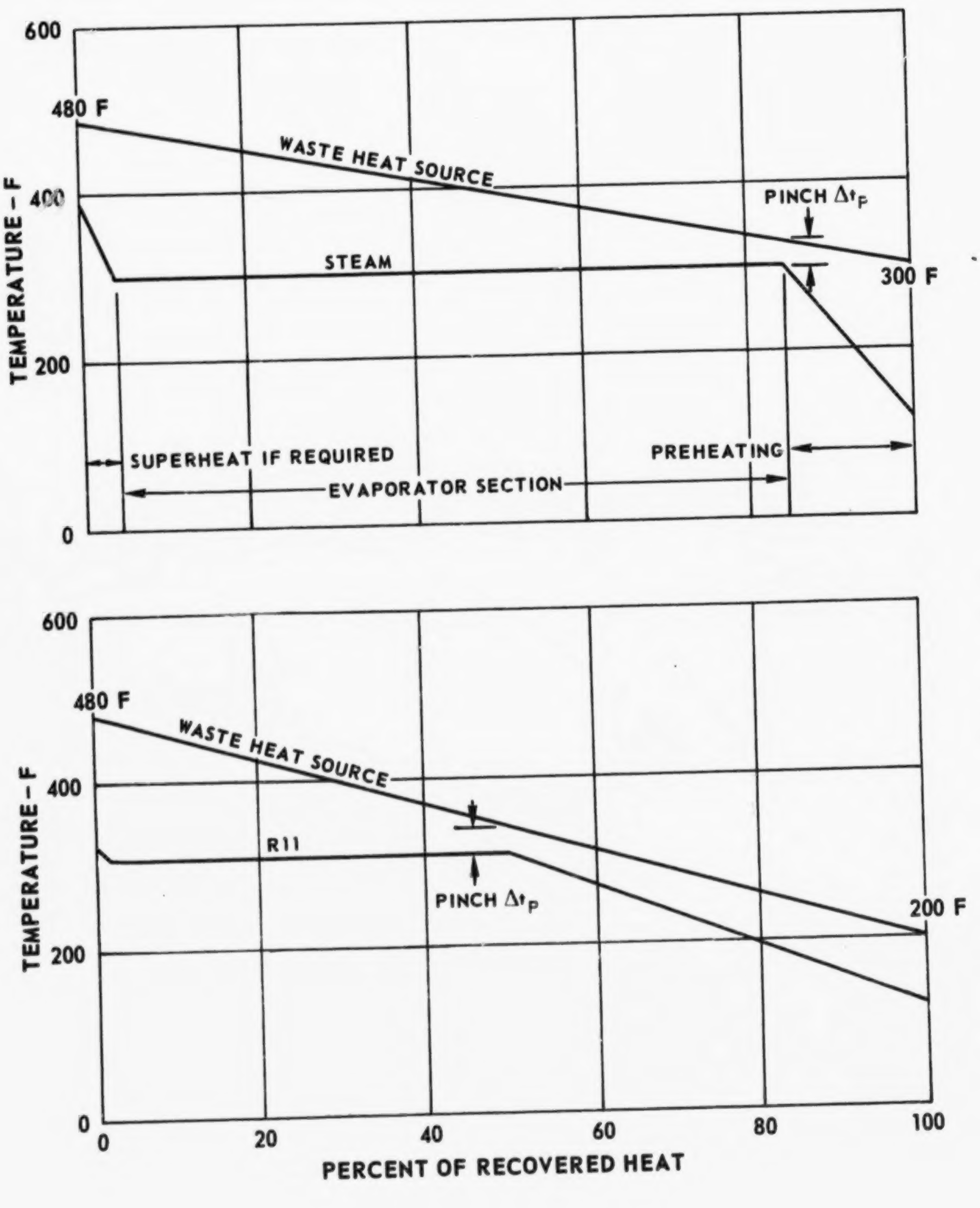


FIG. 95

BOTTOMING CYCLE POWER OUTPUT

MAXIMUM HORSEPOWER AVAILABLE FROM WASTE HEAT

SENSIBLE HEAT CAPACITY RATE OF WASTE HEAT SOURCE, $w c_p$ = 200,000 Btu/hr F
VAPOR GENERATOR PINCH TEMPERATURE DIFFERENCE = 50 F
TURBINE EFFICIENCY = 80%
CONDENSER TEMPERATURE = 120 F

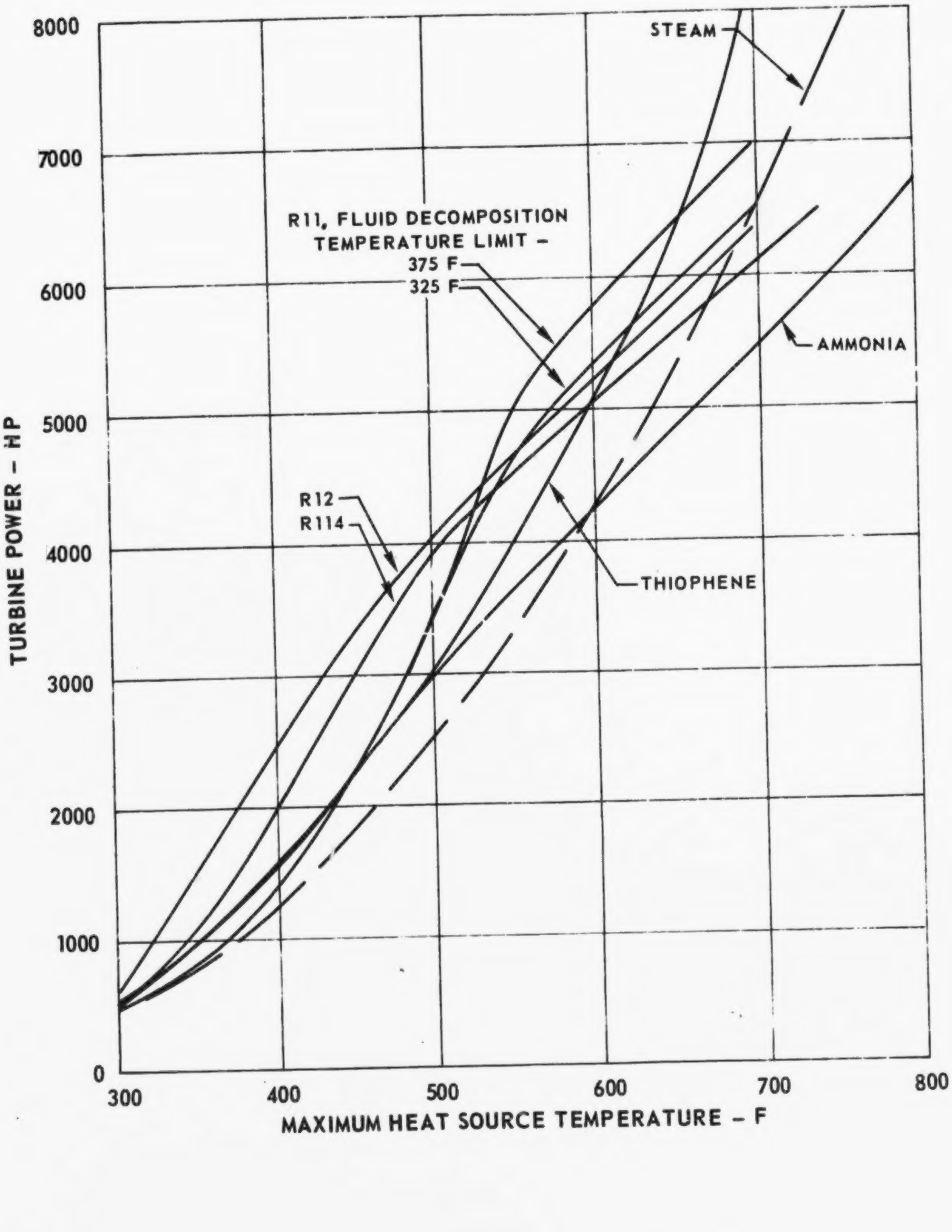
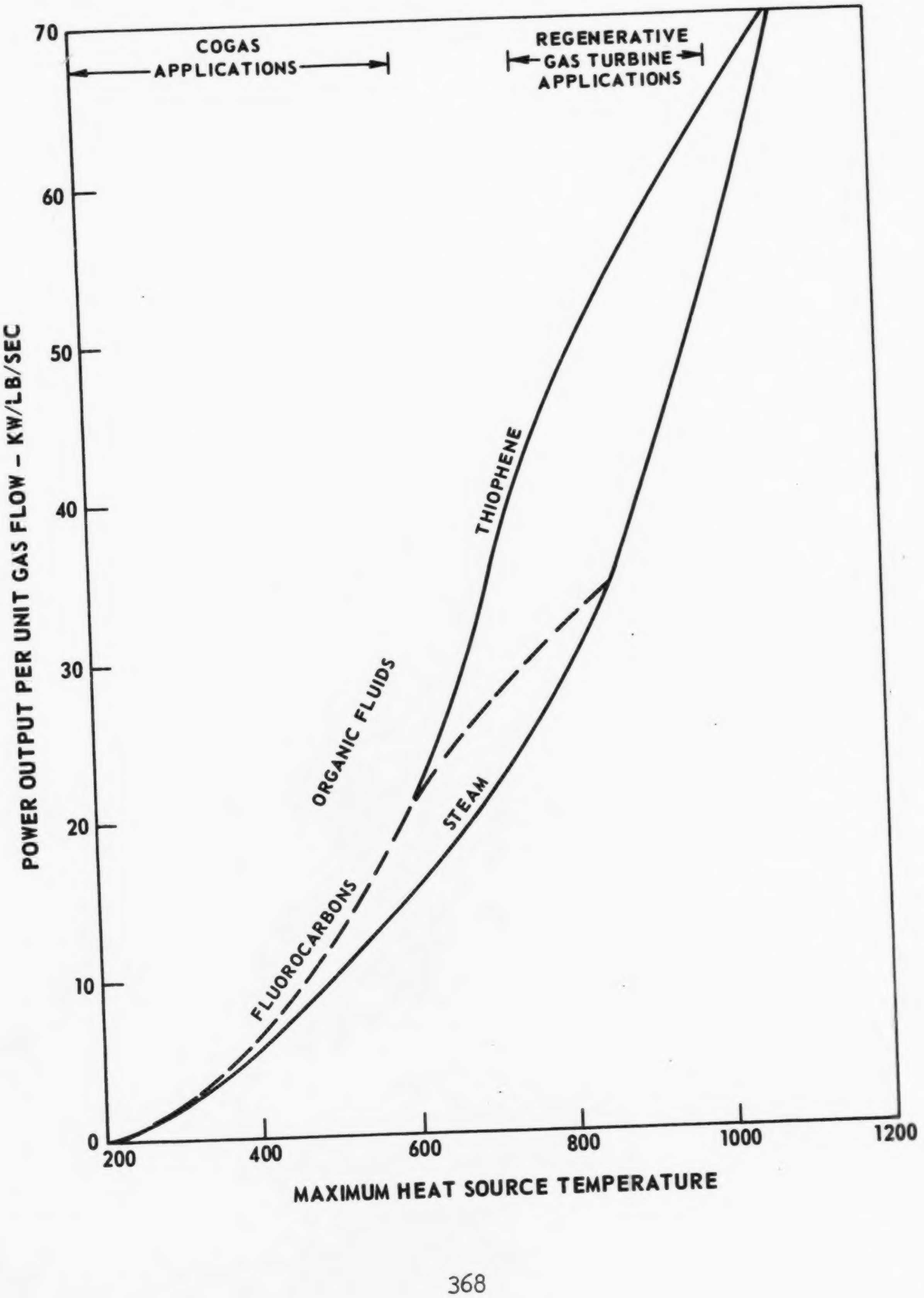


FIG. 96

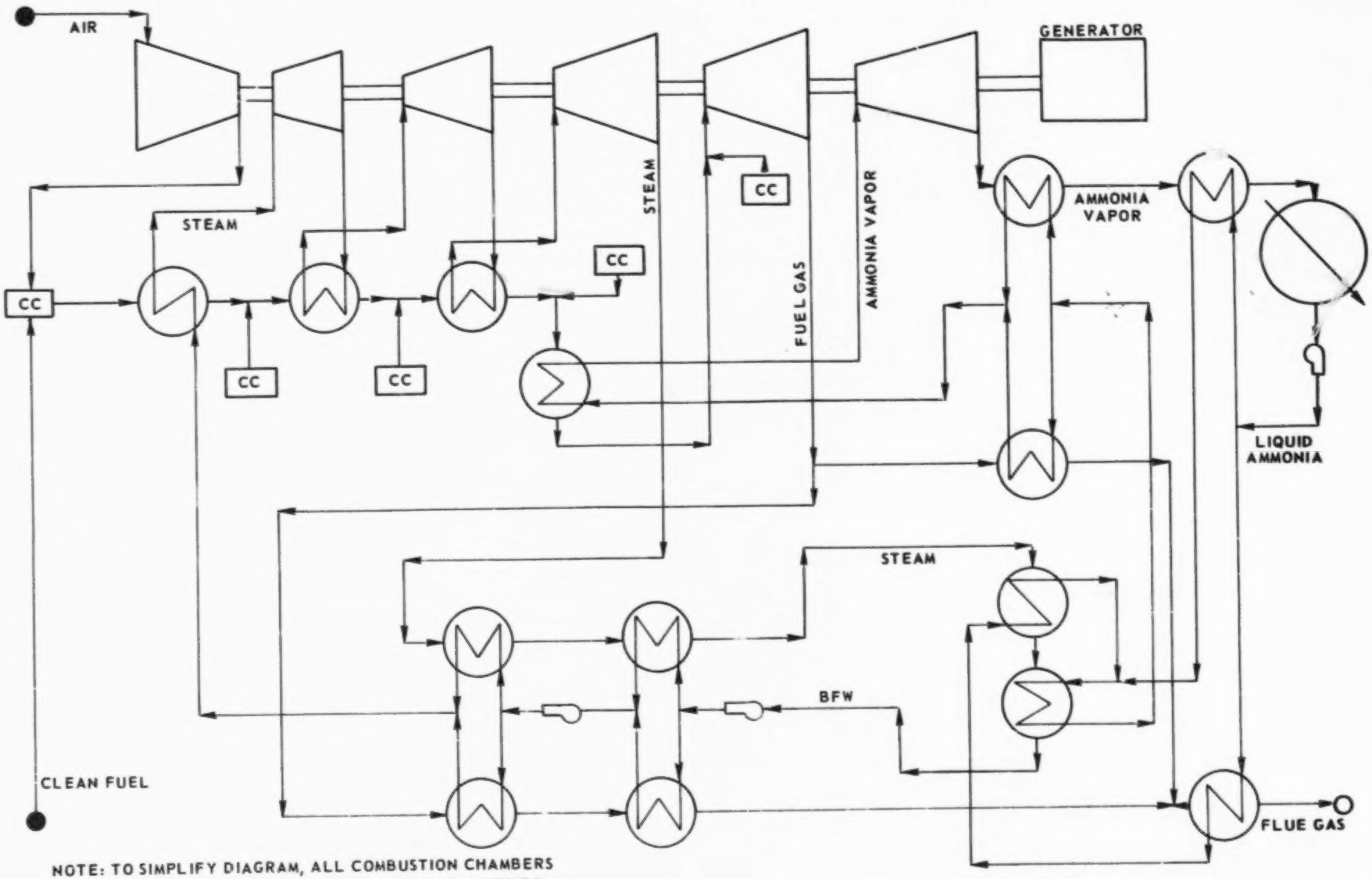
BOTTOMING CYCLE POWER OUTPUT

MAXIMUM AVAILABLE FROM GAS TURBINE STACK GAS
BOILER PINCH TEMPERATURE DIFFERENCE = 50 F

CONDENSER TEMPERATURE = 100 F



TERTIARY POWER CYCLE WITH HIGH-TEMPERATURE HEAT ADDITION TO AMMONIA BOTTOMING CYCLE



NOTE: TO SIMPLIFY DIAGRAM, ALL COMBUSTION CHAMBERS EXCEPT THE FIRST ARE SHOWN WITHOUT LINES TO INDICATE FLOW OF FUEL AND AIR.

FIG. 97

FIG. 98

SIMPLIFIED SCHEMATIC FLOW DIAGRAM OF A TERTIARY POWER CYCLE WITH DIRECT HEAT ADDITION

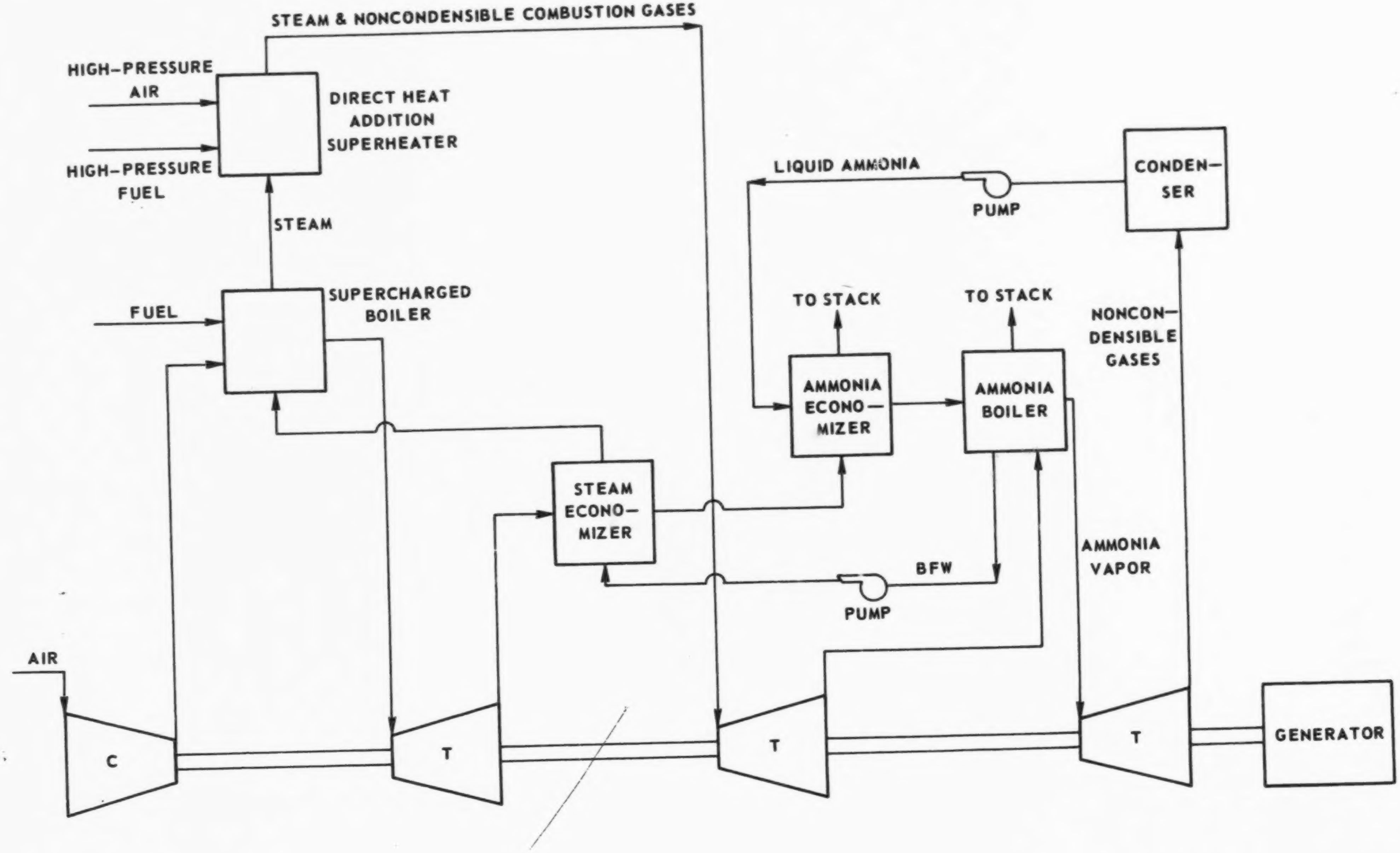


FIG. 99

IDEALIZED CYCLE PERFORMANCE FOR REGENERATIVE GAS TURBINE

TURBINE INLET TEMPERATURE – 1600 F
REGENERATOR EFFECTIVENESS 90%
CYCLE PRESSURE DROP – 5%

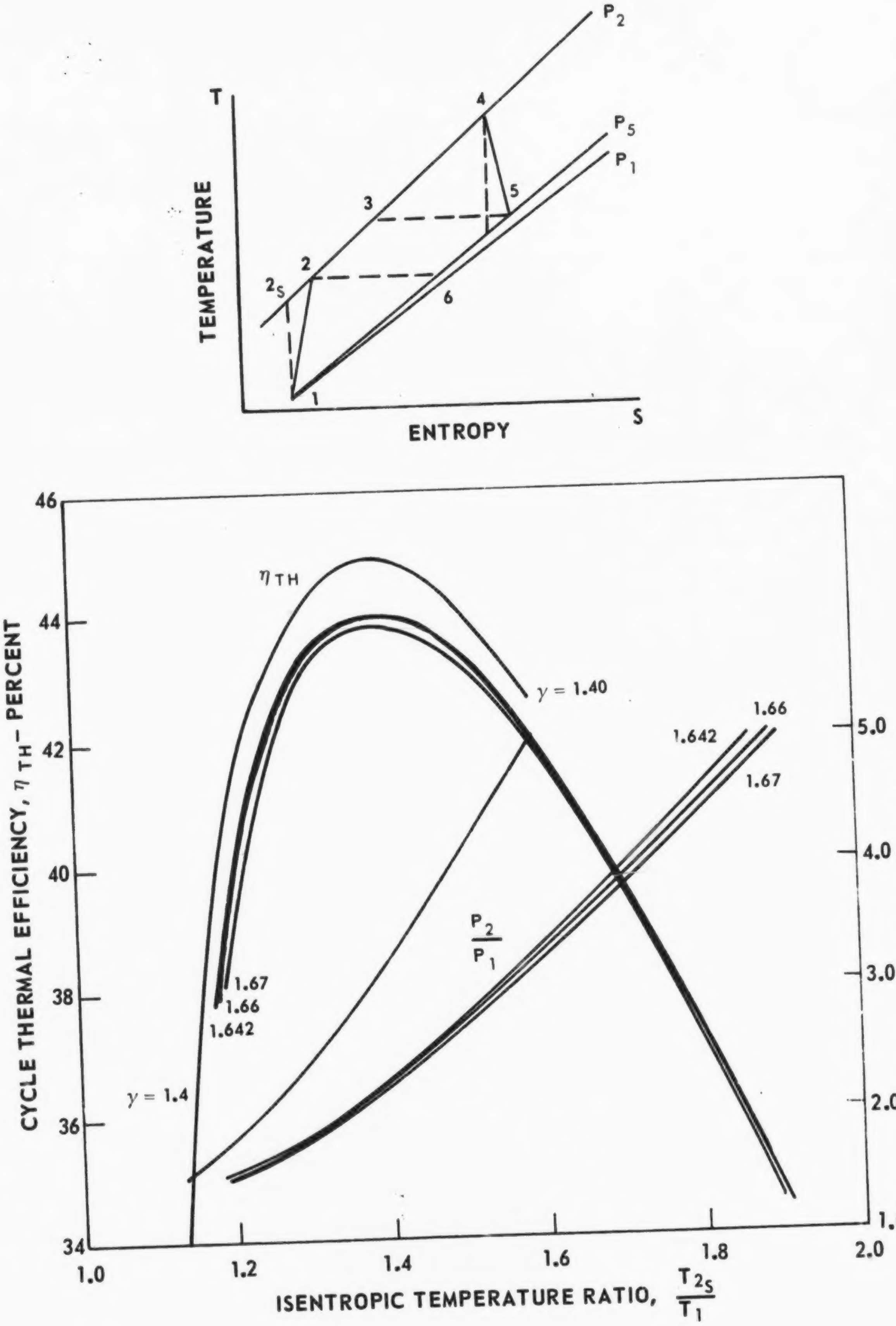


FIG. 100

HEAT TRANSFER CHARACTERISTICS OF GAS COOLANTS FOR CIRCULAR FINNED TUBE SURFACE

GAS TEMPERATURE = 1185 F

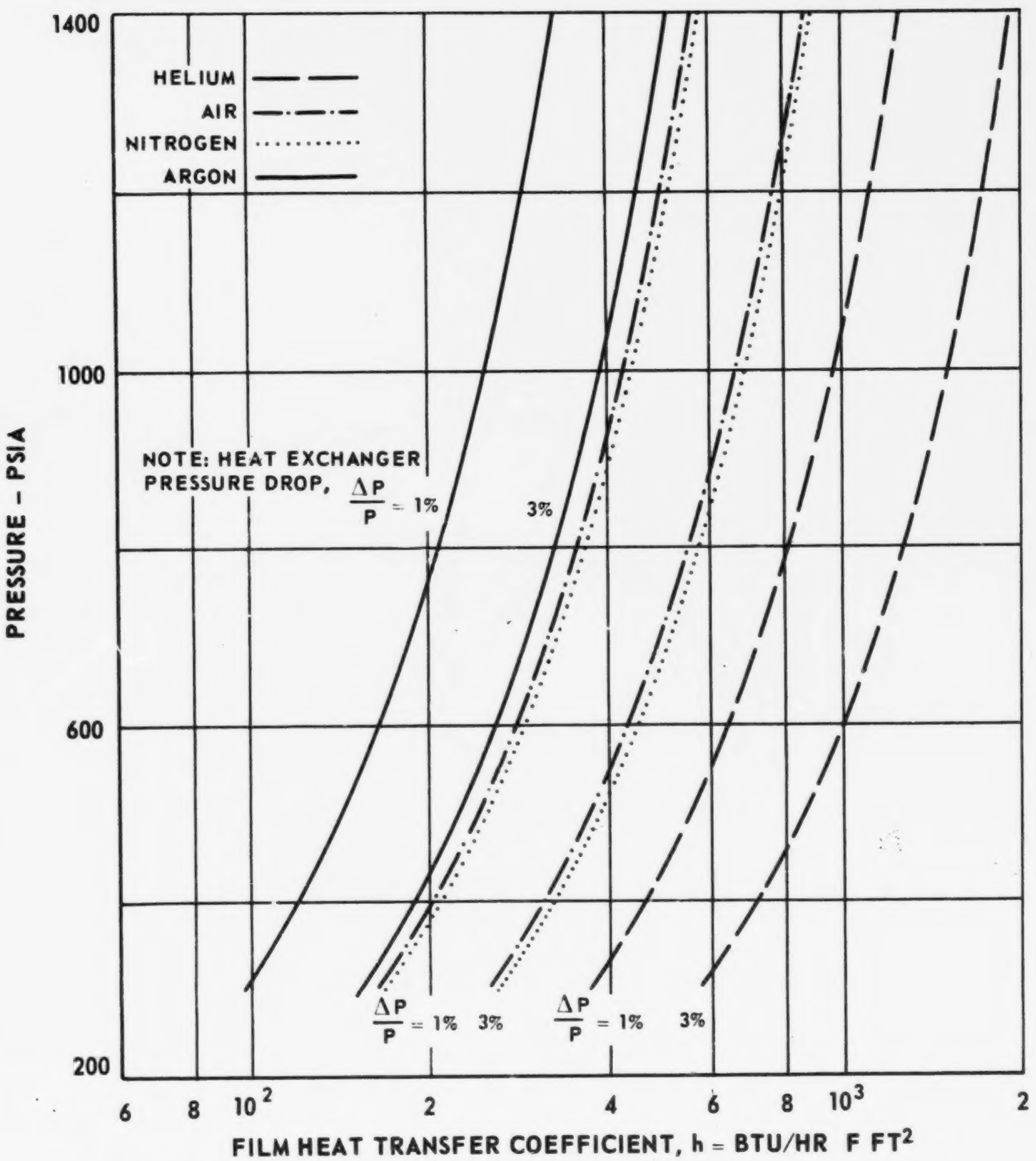


FIG. 101

THE EFFECT OF INTERCOOLING, REGENERATION, AND REHEAT ON THE PERFORMANCE OF A CLOSED-CYCLE GAS TURBINE

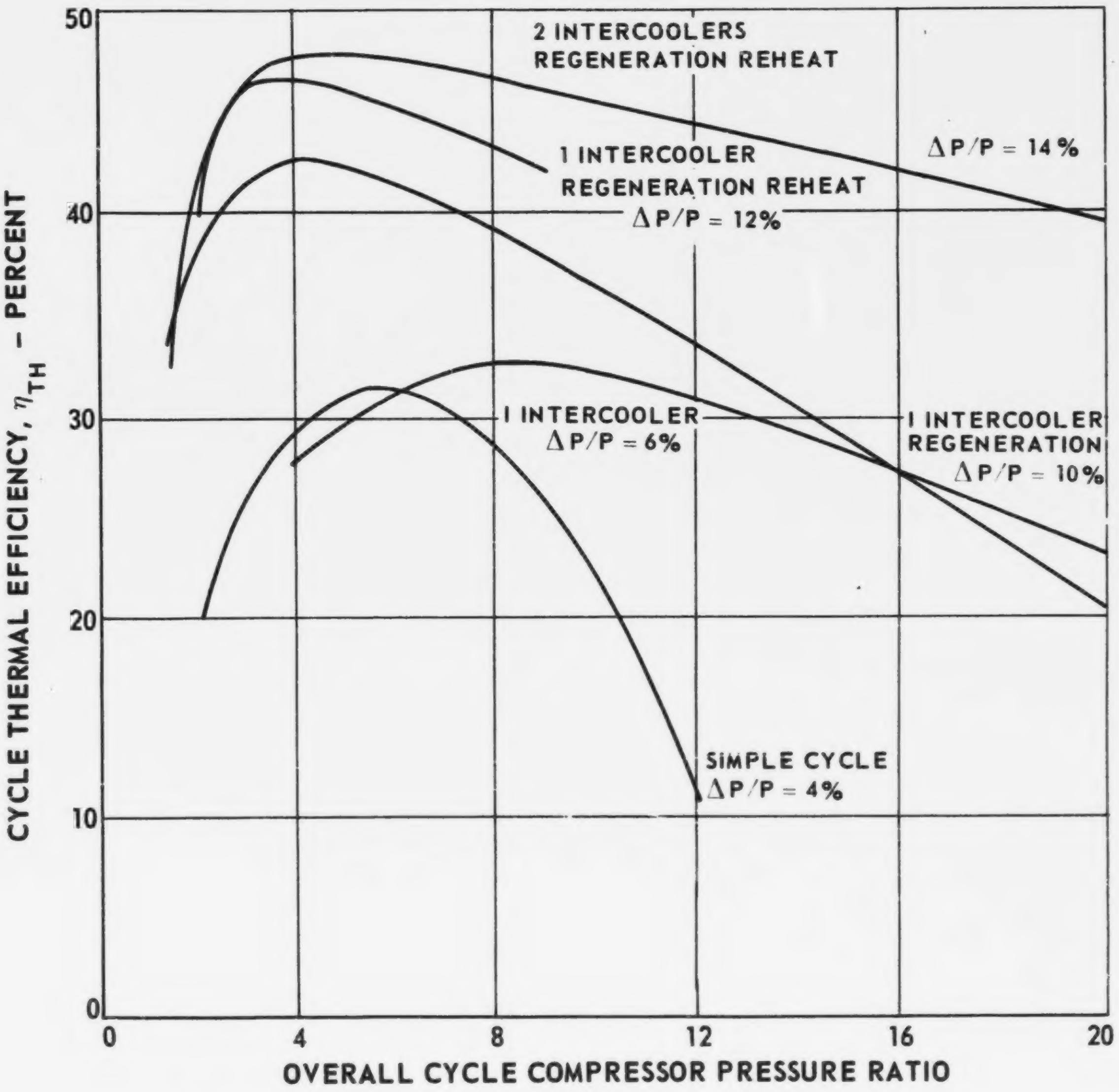
WORKING FLUID – HELIUM

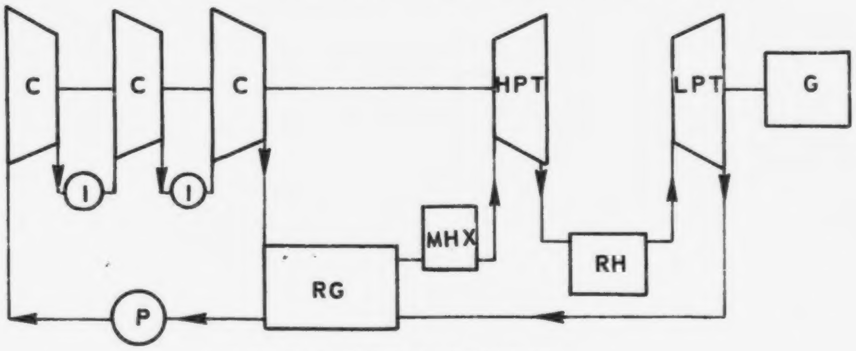
TURBINE INLET TEMPERATURE – 1600 F

COMPRESSOR INLET TEMPEARTURE = 90 F

REGENERATOR EFFECTIVENESS = 90 %

CYCLE PRESSURE DROP, ($\Delta P/P$) - %





TYPICAL ARRANGEMENT

FIG. 102

THE EFFECT OF REGENERATOR EFFECTIVENESS ON THE PERFORMANCE OF A CLOSED-CYCLE GAS TURBINE

WORKING FLUID – HELIUM
CYCLE PRESSURE DROP = 5%
TURBINE INLET TEMPERATURE = 1600 F
COMPRESSOR INLET TEMPERATURE = 90 F
REGENERATOR EFFECTIVENESS = ξ_r – PERCENT

NO REHEAT

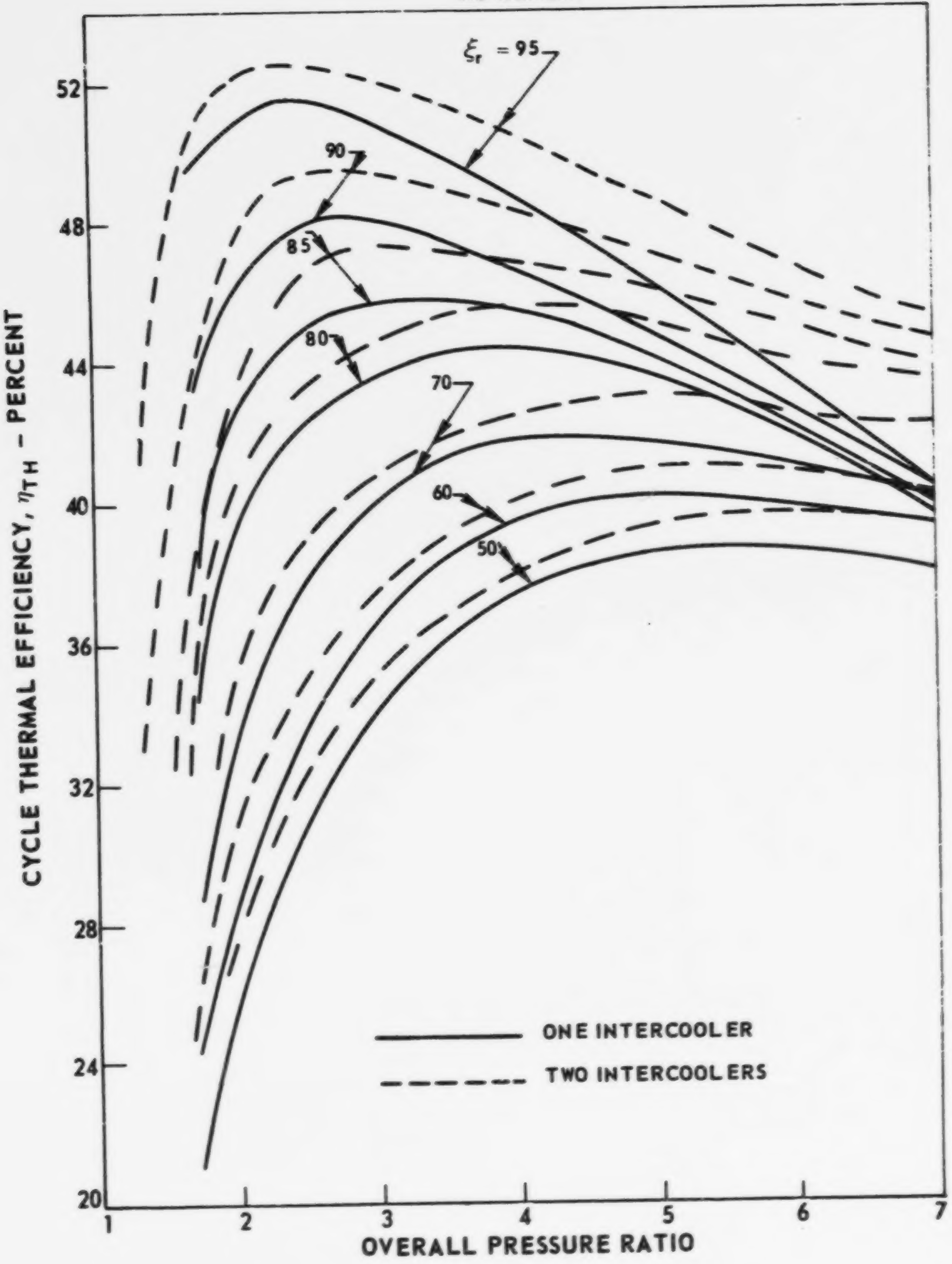


FIG. 103

CLOSED-CYCLE HELIUM TURBINE PERFORMANCE

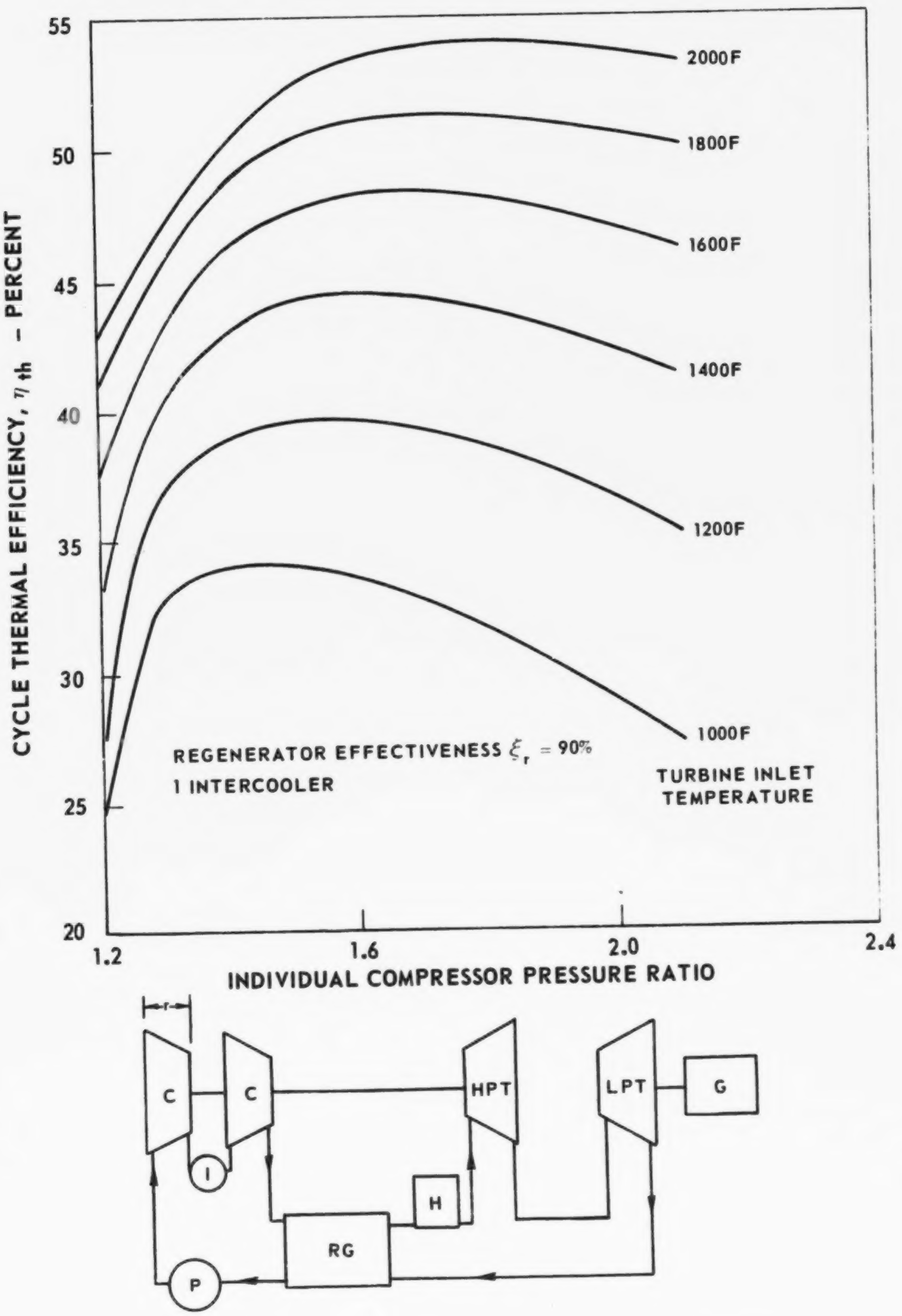
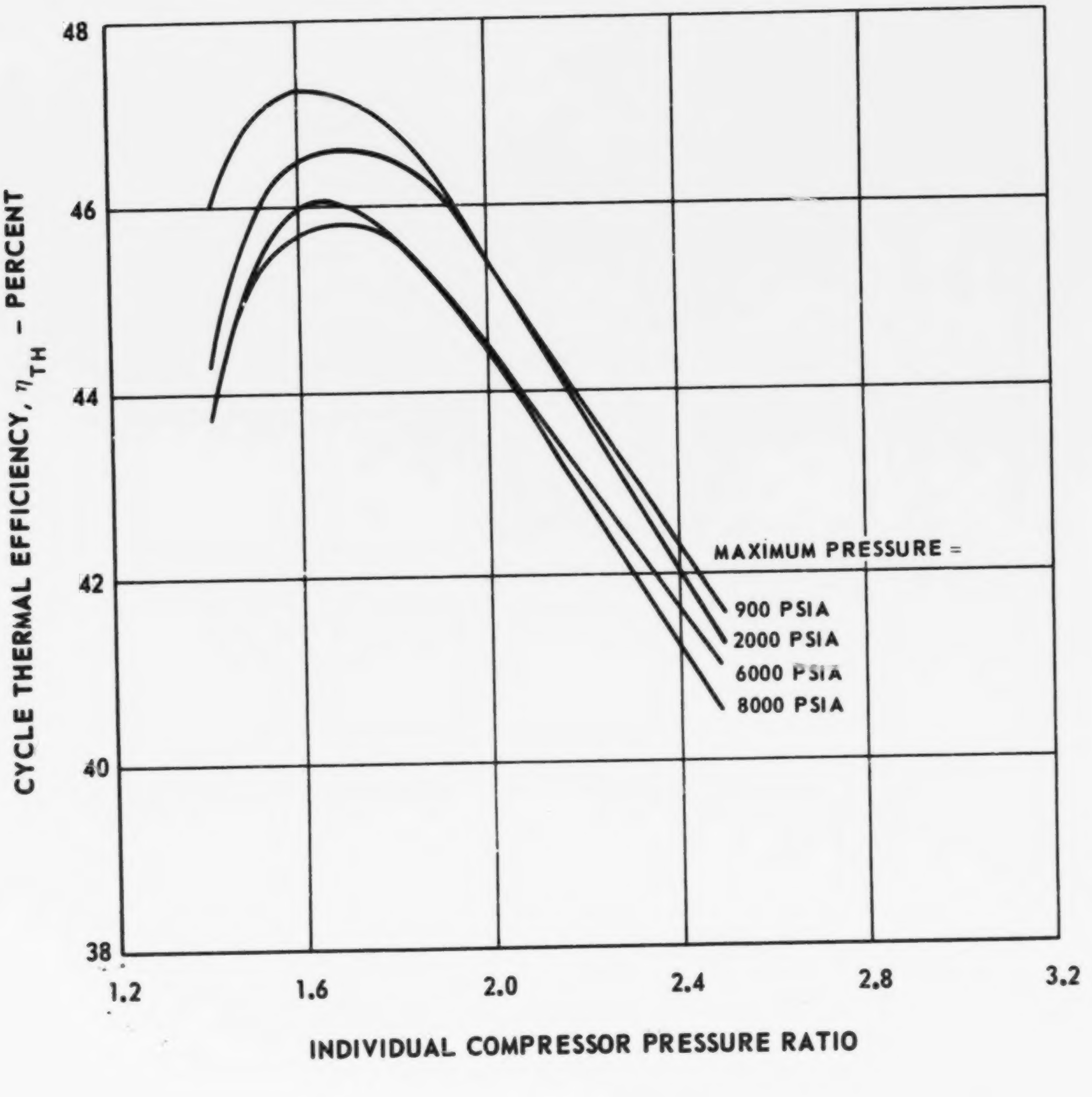


FIG. 104

THE EFFECT OF SYSTEM PRESSURIZATION LEVEL ON CLOSED-CYCLE GAS TURBINE PERFORMANCE



WORKING FLUID–HELIUM
TURBINE INLET TEMPERATURE = 1600 F
COMPRESSOR INLET TEMPERATURE = 90 F
REGENERATOR EFFECTIVENESS = 90%
CYCLE PRESSURE DROP = 5%
1 INTERCOOLER
NO REHEAT

FIG. 105

EFFECT OF REGENERATOR EFFECTIVENESS ON OPTIMUM REGENERATOR SIZE

WORKING FLUID – HELIUM
TURBINE INLET TEMPERATURE = 1600 F
CORE PRESSURE DROP, $(\Delta P/P) = (\Delta P/P)_H + (\Delta P/P)_C = 2.0\%$

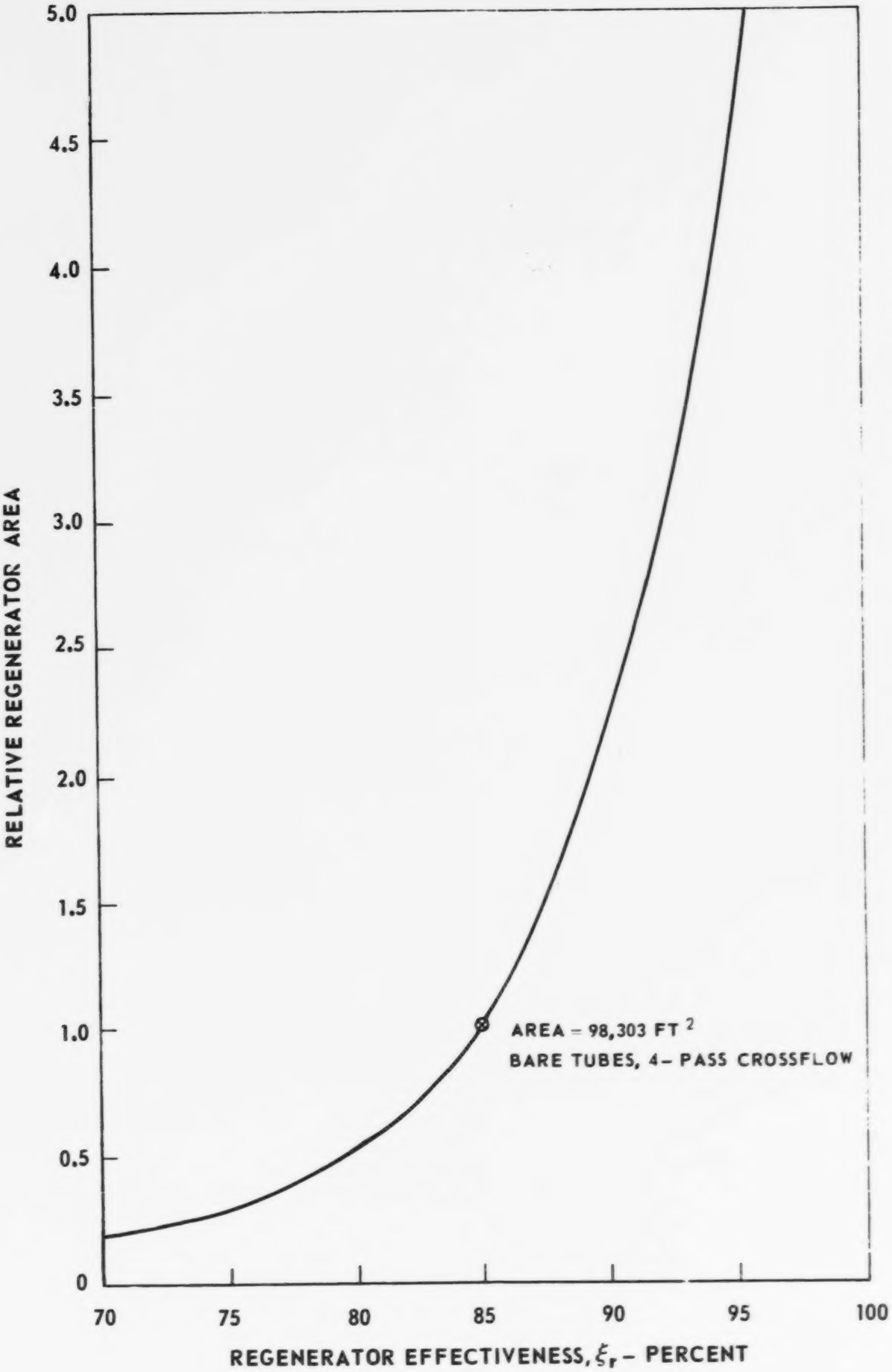
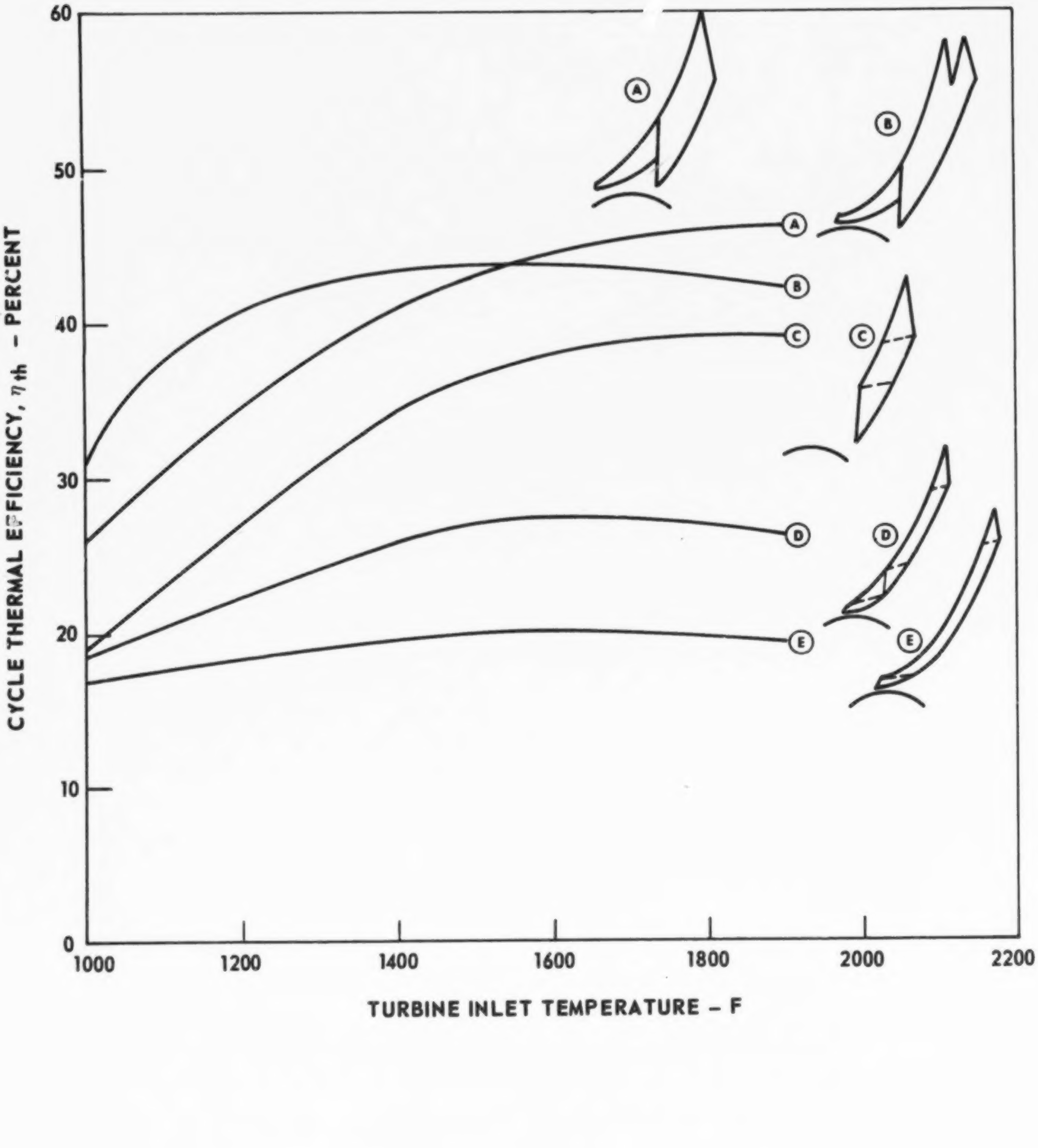


FIG. 106

SUMMARY OF THE PERFORMANCE OF FIVE CARBON DIOXIDE POWER CYCLES



THE EFFECT OF REHEAT AND REGENERATOR EFFECTIVENESS ON THE PERFORMANCE OF THE FEHER-BRAYTON CARBON DIOXIDE CYCLE

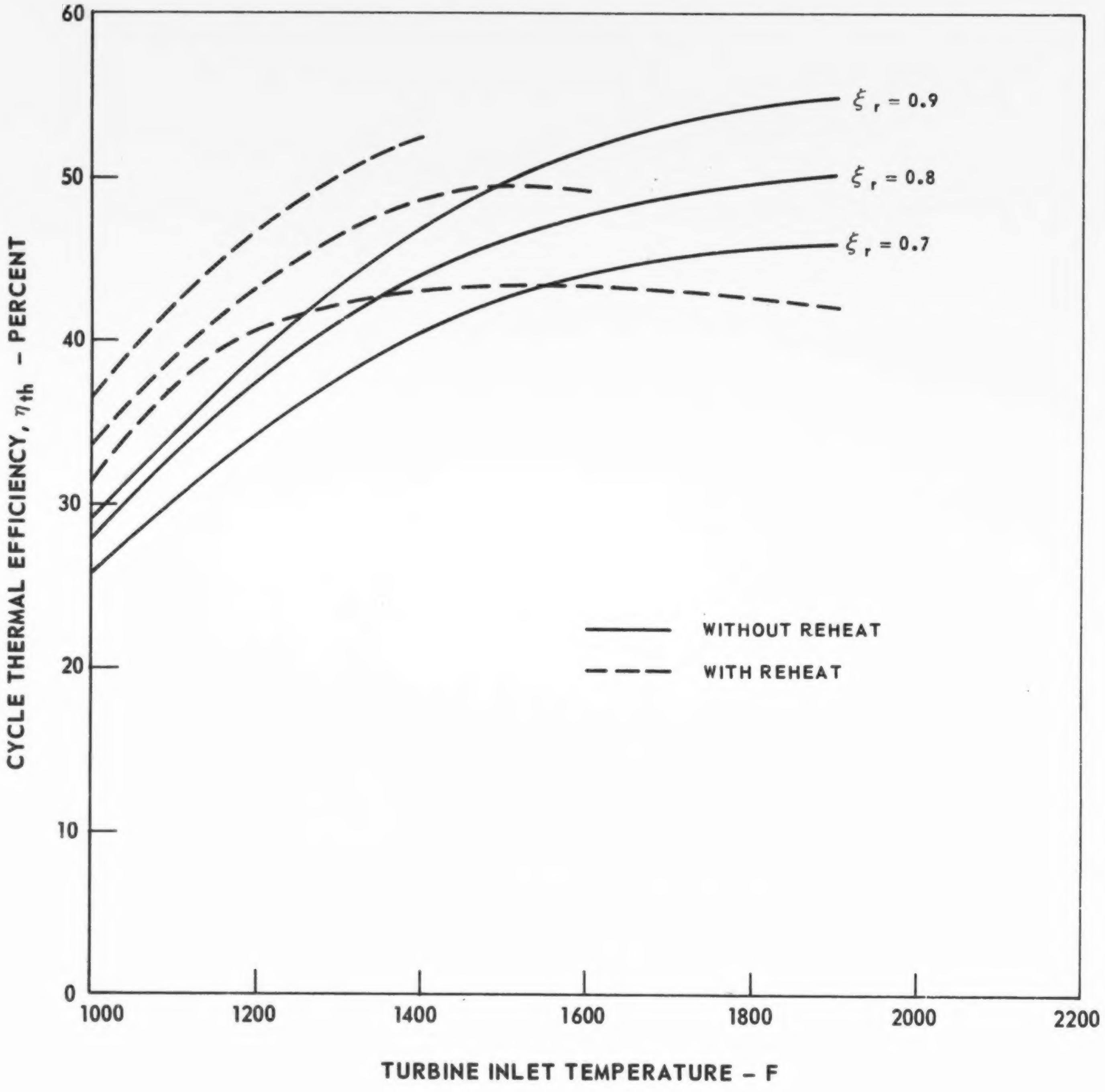


FIG. 108

COMPARISON OF SULFUR DIOXIDE CYCLE PERFORMANCE

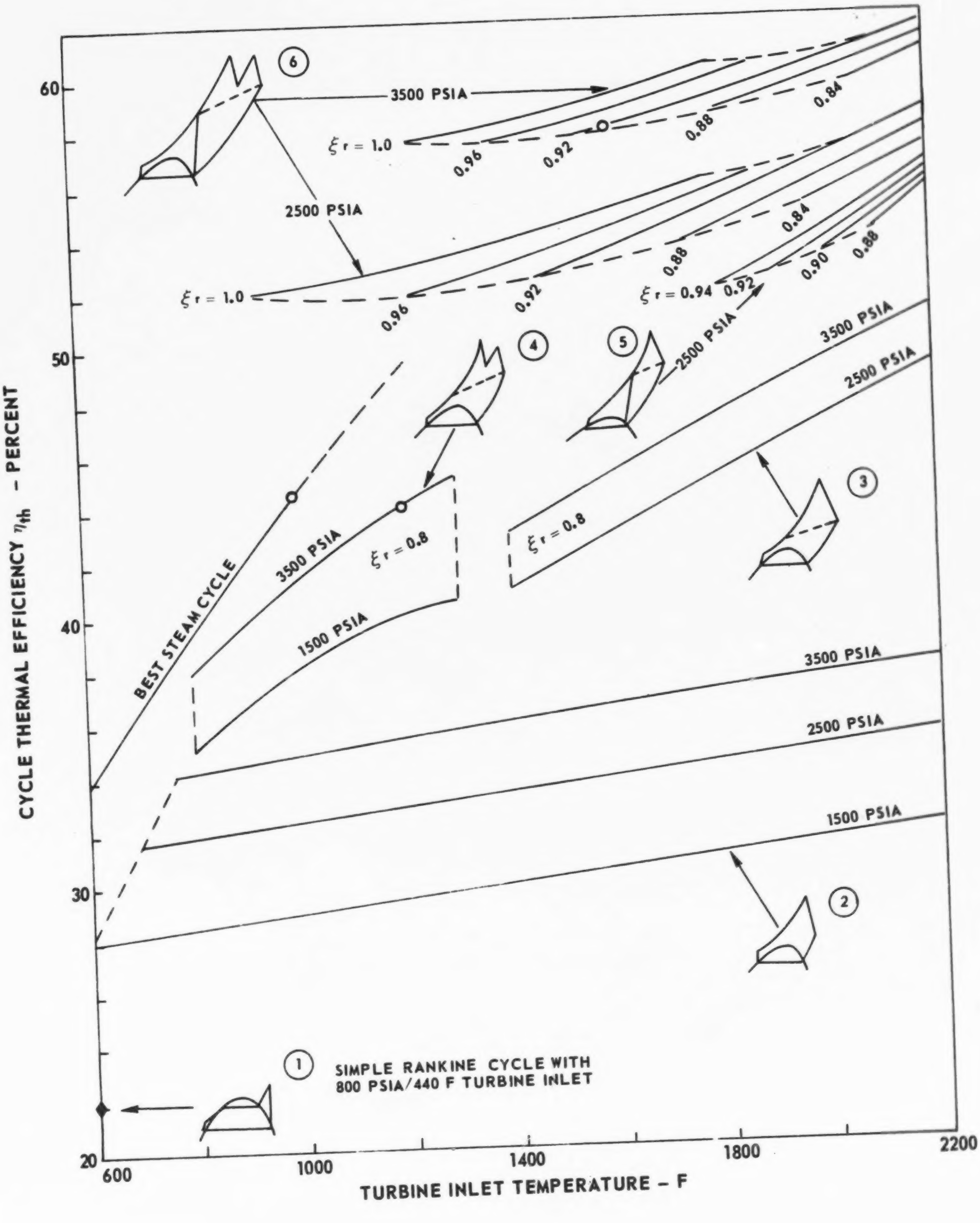


FIG. 109

PERFORMANCE OF SUPERCRITICAL SULFUR DIOXIDE CYCLE WITH REHEAT AND REGENERATION

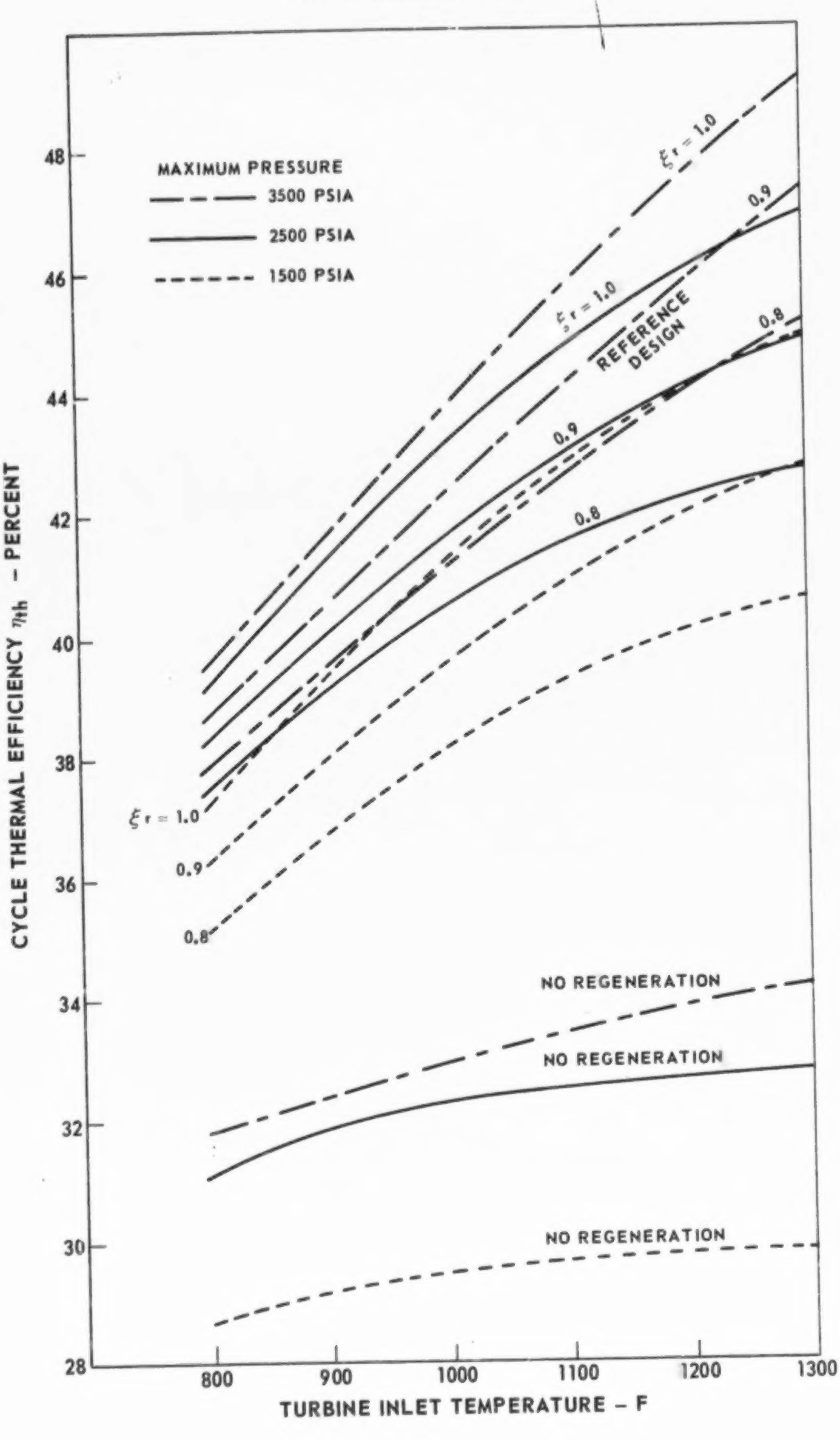


FIG. 110

PERFORMANCE OF PARTIAL-CONDENSATION SULFUR DIOXIDE CYCLE WITH REHEAT AND REGENERATION

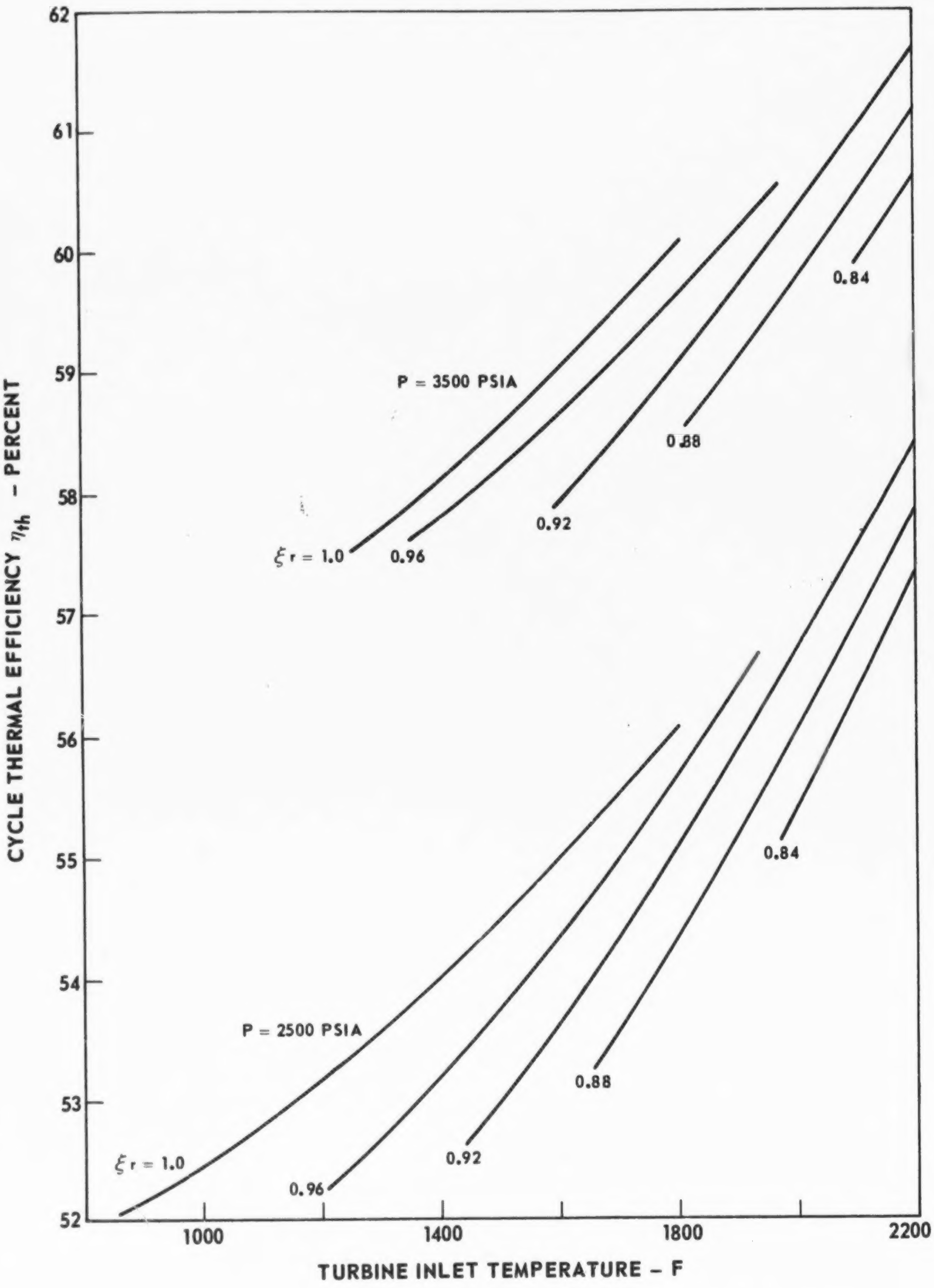


FIG. 111

TYPICAL DISTRIBUTIONS OF HEAT FLUX, TUBE WALL TEMPERATURE, AND WORKING FLUID TEMPERATURE IN A VERTICAL FIRED FLUID HEATER

$Q_M = 3 \times 10^4$ BTU/FT^2 – HR

$T_{FM} = 1112$ F

$T_{W_{MAX}} = 1398$ F

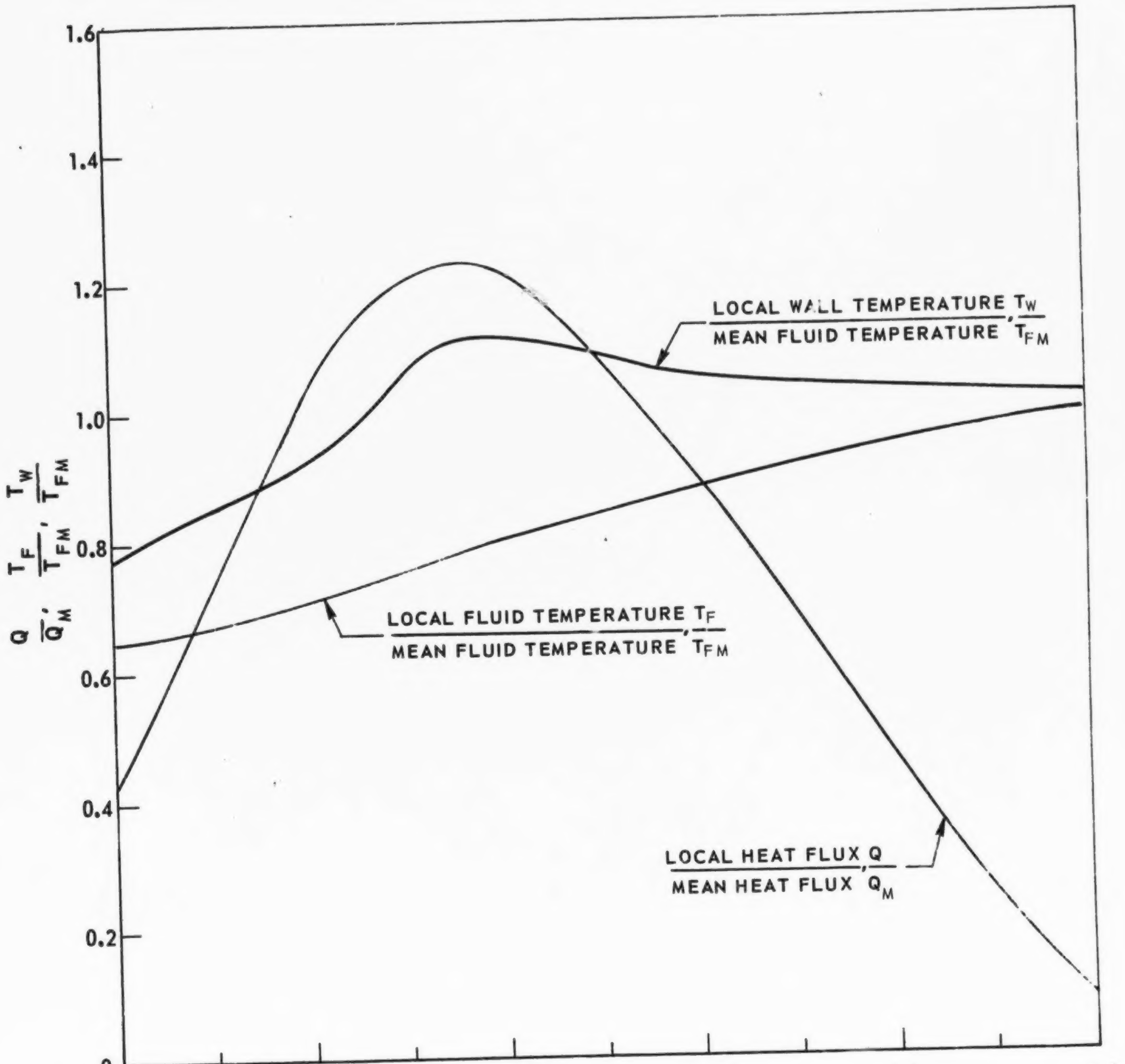


FIG. 112

MAXIMUM ALLOWABLE STRESS FOR SELECTED HEAT EXCHANGER MATERIAL

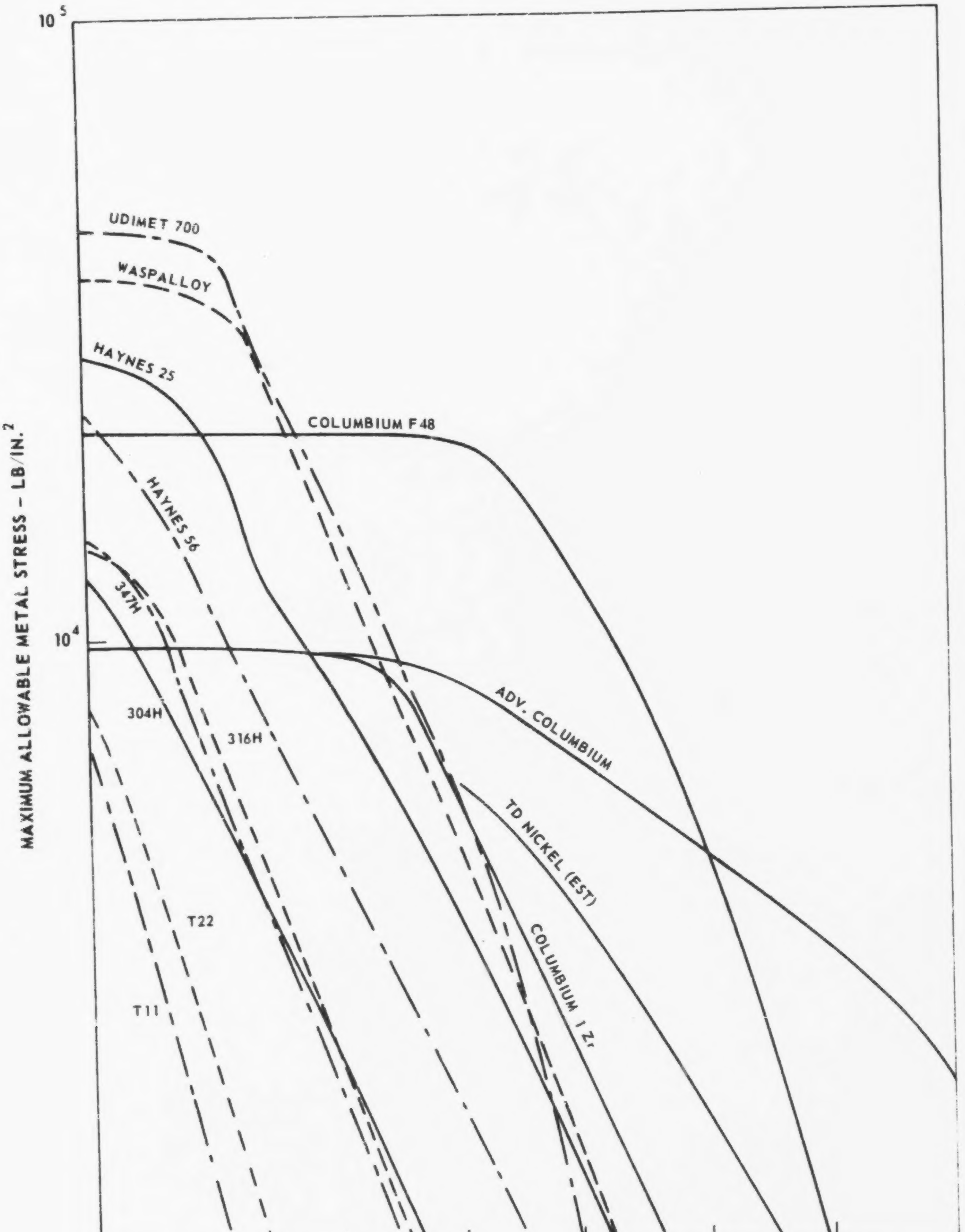


FIG. 113

PUBLISHED CORROSION RATES OF VARIOUS ALLOYS IN CARBON DIOXIDE

METAL TEMPERATURE = 1652 F

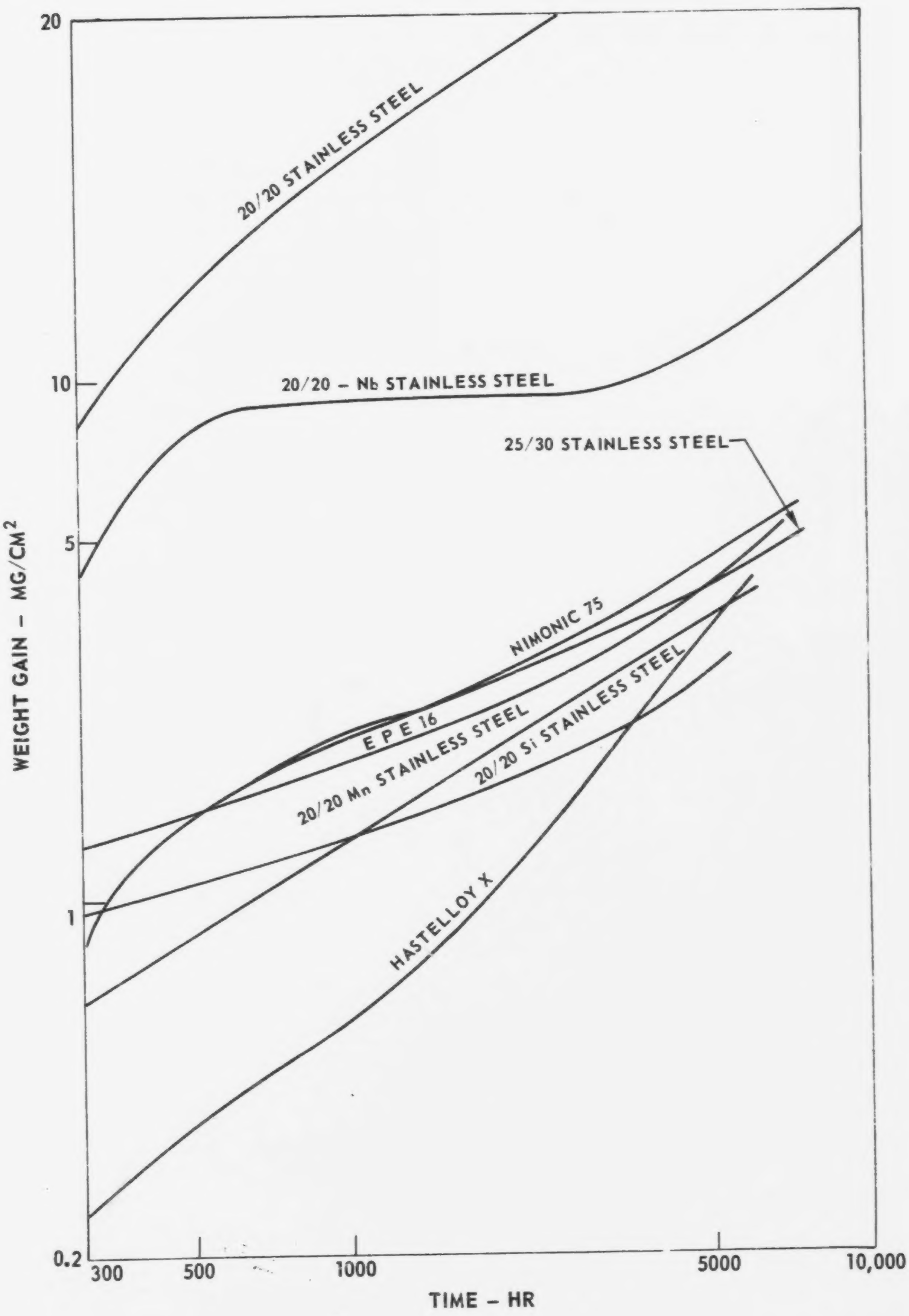


FIG. 114

HELIUM REGENERATOR DESIGN CHARACTERISTICS

CORE PRESSURE DROP $(\Delta P/P)_T = (\Delta P/P)_H + (\Delta P/P)_C = 2.0\%$ ————

$= 1.0\%$ — — —

EXCHANGER EFFECTIVENESS = 85%

EXCHANGER SURFACE – BARE TUBES, 6 PASS – CROSSFLOW

MATERIAL – CARBON STEEL

* VALUES ARE NONDIMENSIONAL

$A_{TH}^* = A_{TH}/A_{TH_{MIN}}$, $A_H^* = A_H/A_{H_{MIN}}$, $V^* = V/V_{MIN}$ $Z^* = Z/Z_{MIN}$

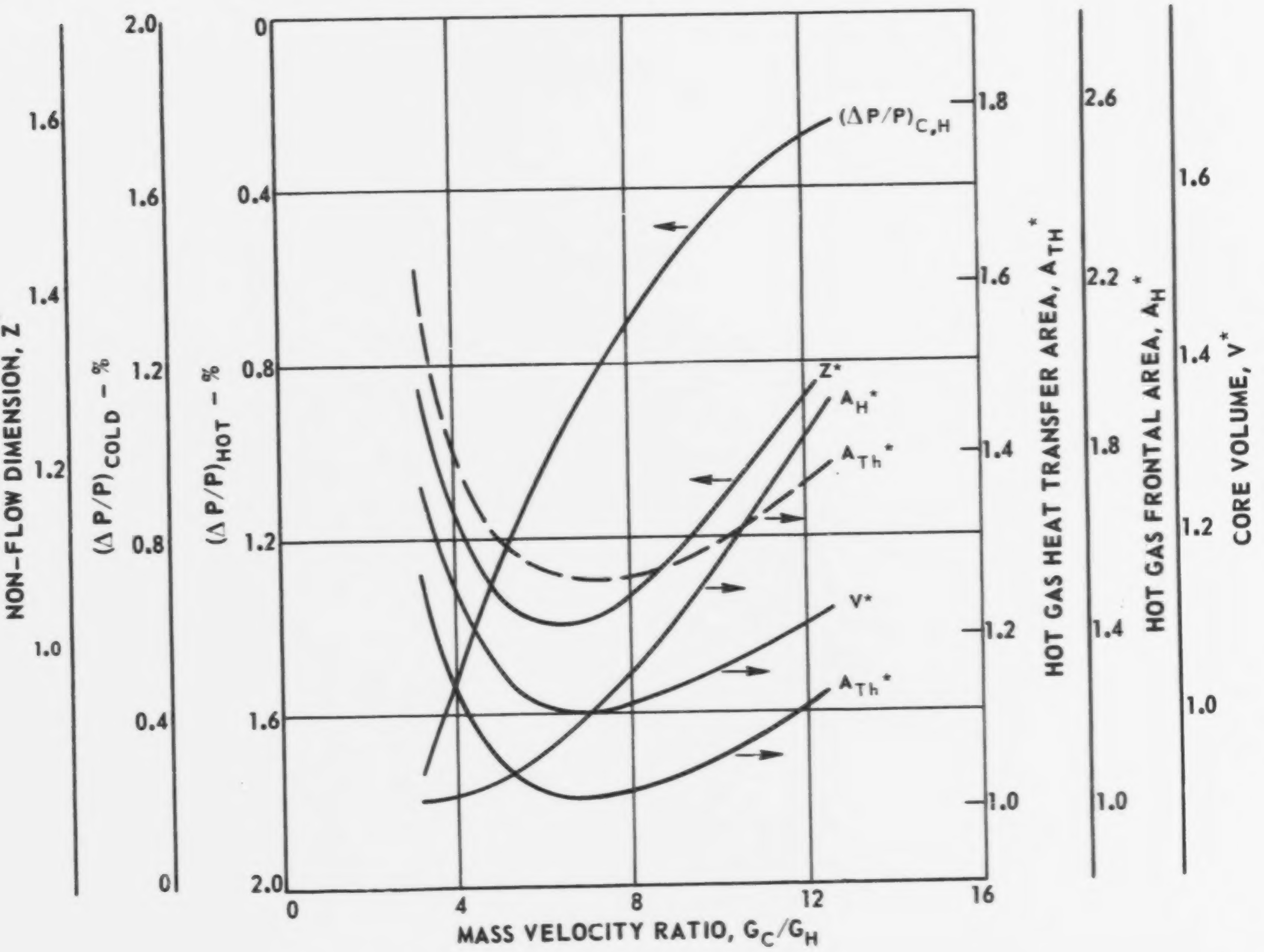


FIG. 115

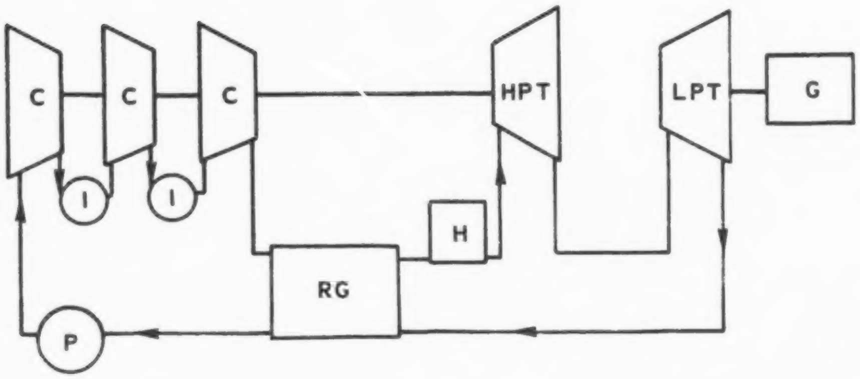
CONFIGURATION OF HELIUM CLOSED-CYCLE GAS TURBINE POWER SYSTEMS

$T_{MAX} = 1600$ F

$\xi_r = 90\%$

$\Delta P/P = 6.5\%$

$\eta_{th} = 48.5\%$



$T_{MAX} = 1200$ F

$\xi_r = 92.5\%$

$\Delta P/P = 9\%$

$\eta_{th} = 42.3\%$

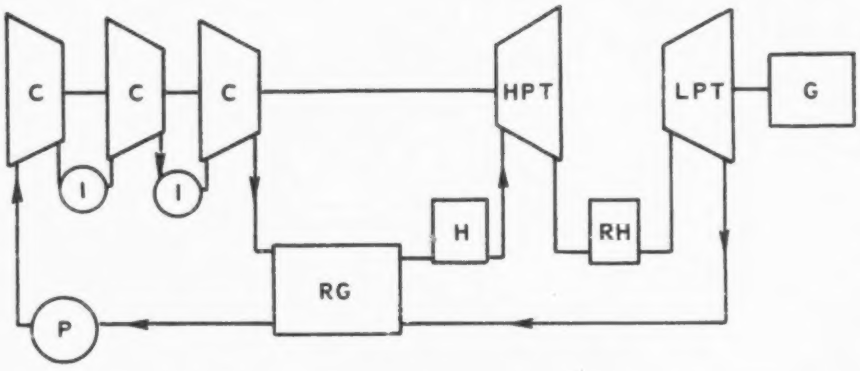


FIG. 116

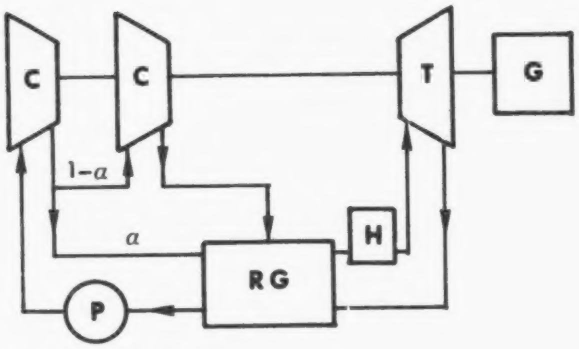
CONFIGURATION OF CARBON DIOXIDE POWER SYSTEMS

T_{MAX} = 1600 F

ξ_r = 70%

$\Delta P/P$ = 15%

η_{th} = 44.7%



T_{MAX} = 1200 F

ξ_r = 95%

$\Delta P/P$ = 15%

η_{th} = 44.6%

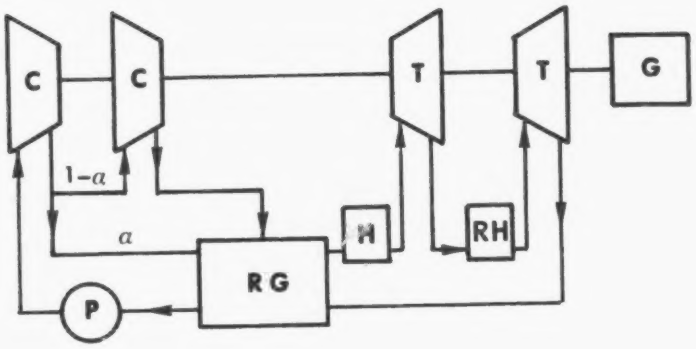


FIG. 117

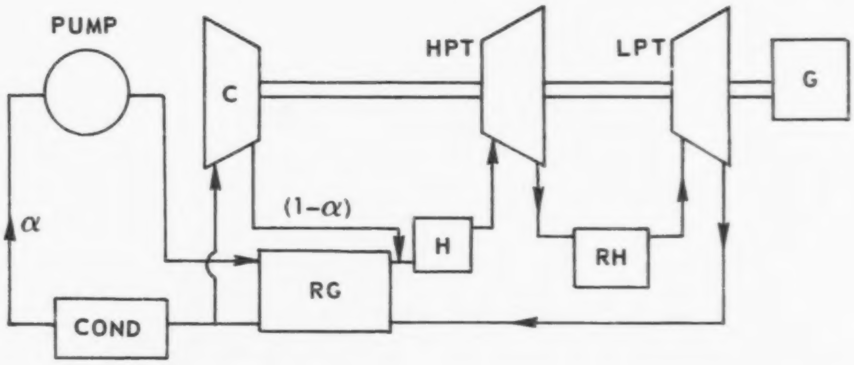
CONFIGURATION OF SULFUR DIOXIDE POWER SYSTEMS

$T_{MAX} = 1600$ F

$\xi_r = 92\%$

$\Delta P/P = 15\%$

$\eta_{th} = 57.5\%$



$T_{MAX} = 1200$ F

$\xi_r = 80\%$

$\Delta P/P = 15\%$

$\eta_{th} = 42.5\%$

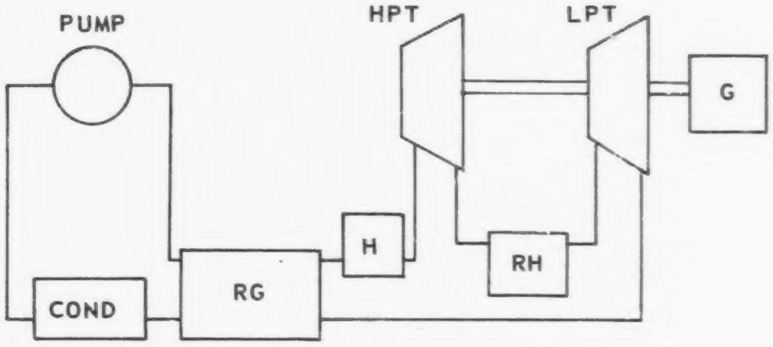


FIG. 118

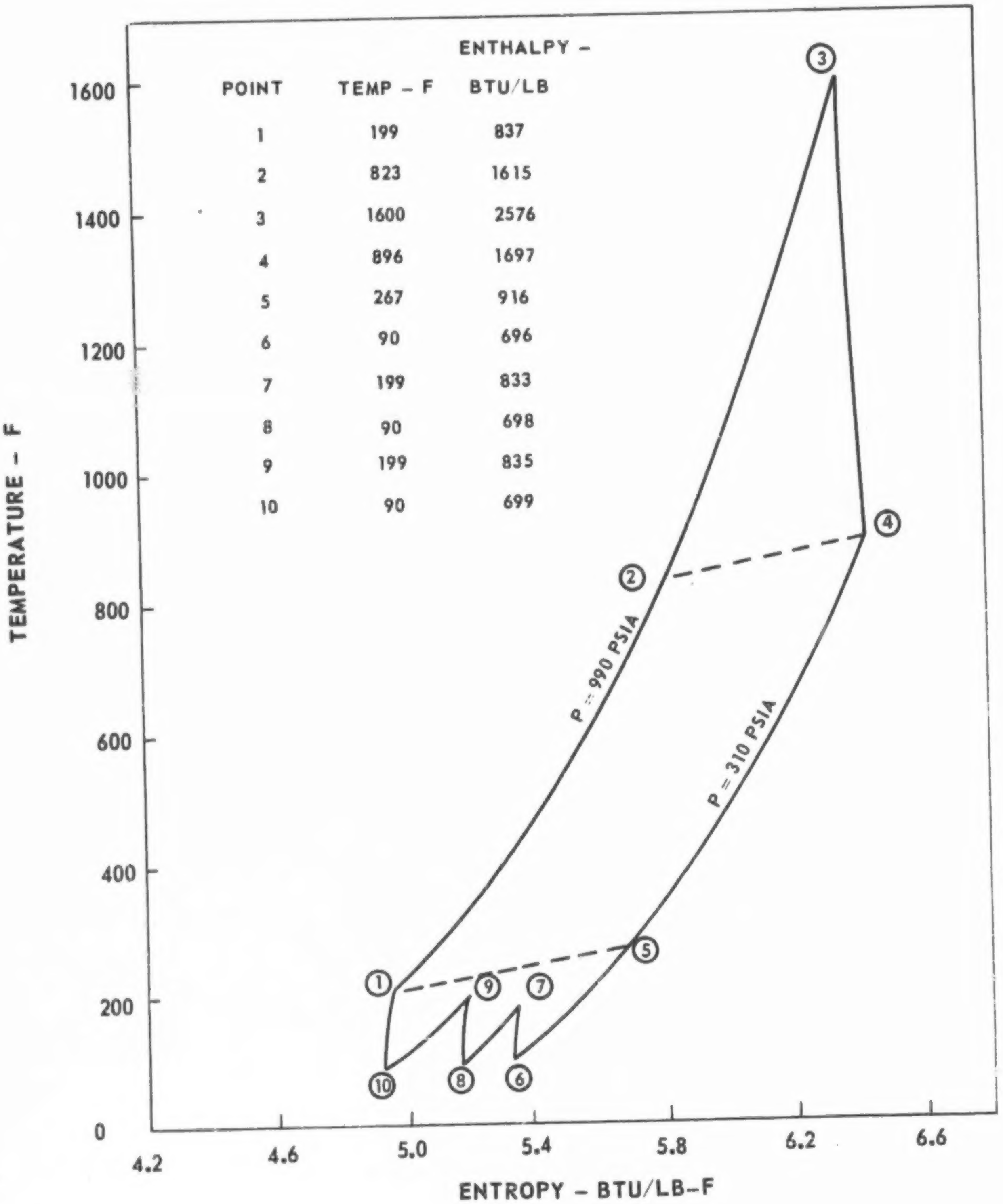
HIGH-TEMPERATURE CLOSED-CYCLE HELIUM TURBINE WITH INTERCOOLING AND REGENERATION

$T_{MAX} = 1600$ F

$\xi_r = 90\%$

$\frac{\Delta P}{P} = 6.5\%$

$\eta_{th} = 48.5\%$



POINT	TEMP – F	ENTHALPY – BTU/LB
1	199	837
2	823	1615
3	1600	2576
4	896	1697
5	267	916
6	90	696
7	199	833
8	90	698
9	199	835
10	90	699

FIG. 119

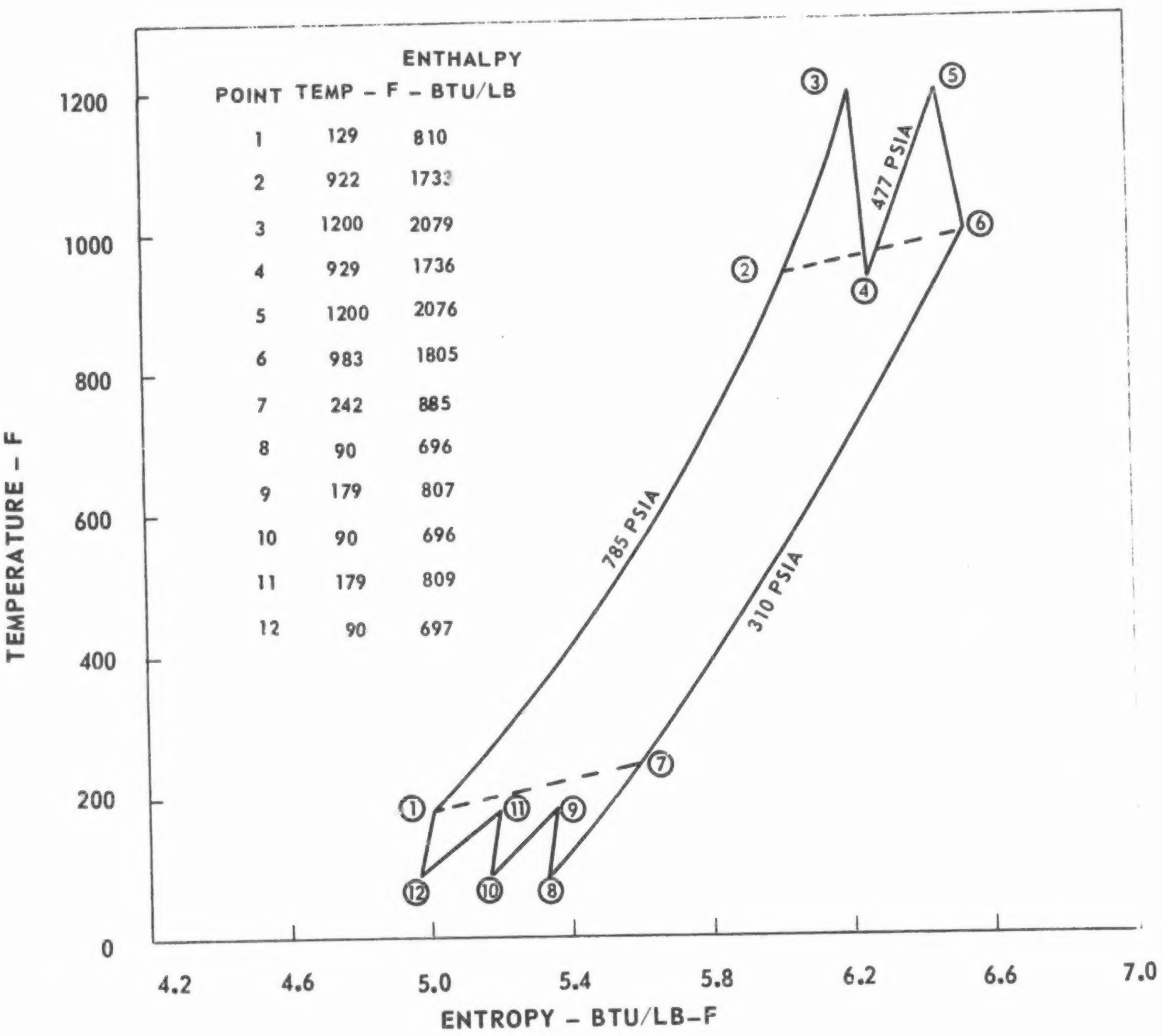
LOW-TEMPERATURE CLOSED-CYCLE HELIUM TURBINE WITH INTERCOOLING REGENERATION AND REHEAT

$T_{MAX} = 1200F$

$\xi_r = 92.5\%$

$\frac{\Delta P}{P} = 9\%$

$\eta_{th} = 42.3\%$



POINT	TEMP - F	ENTHALPY - BTU/LB
1	129	810
2	922	1733
3	1200	2079
4	929	1736
5	1200	2076
6	983	1805
7	242	885
8	90	696
9	179	807
10	90	696
11	179	809
12	90	697

FIG. 120

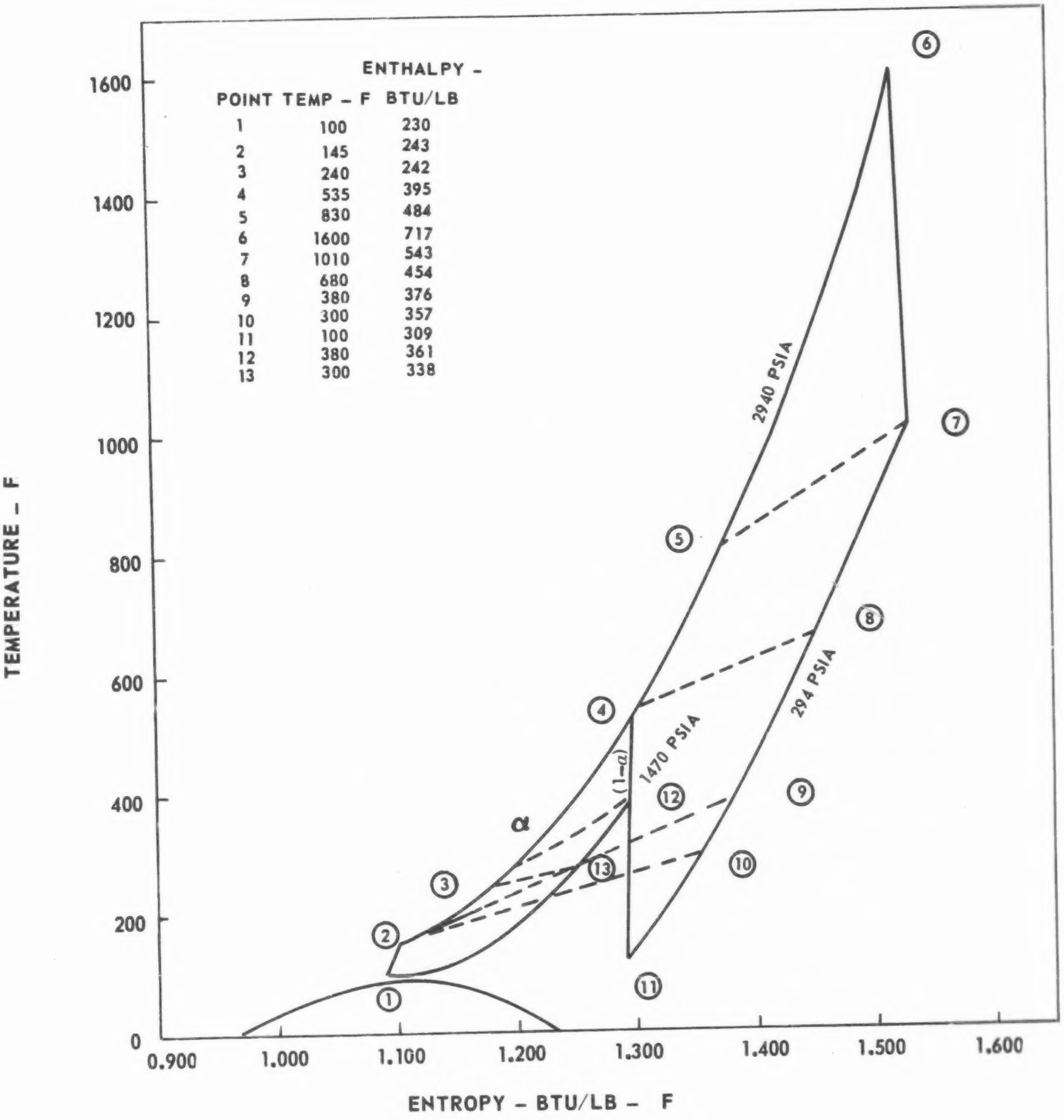
HIGH-TEMPERATURE CO_2 CYCLE

$T_{MAX} = 1600$ F

$\xi_r = 70\%$

$\frac{\Delta P}{P} = 15\%$

$\eta_{th} = 44.7\%$



POINT	TEMP – F	ENTHALPY – BTU/LB
1	100	230
2	145	243
3	240	242
4	535	395
5	830	484
6	1600	717
7	1010	543
8	680	454
9	380	376
10	300	357
11	100	309
12	380	361
13	300	338

FIG. 121

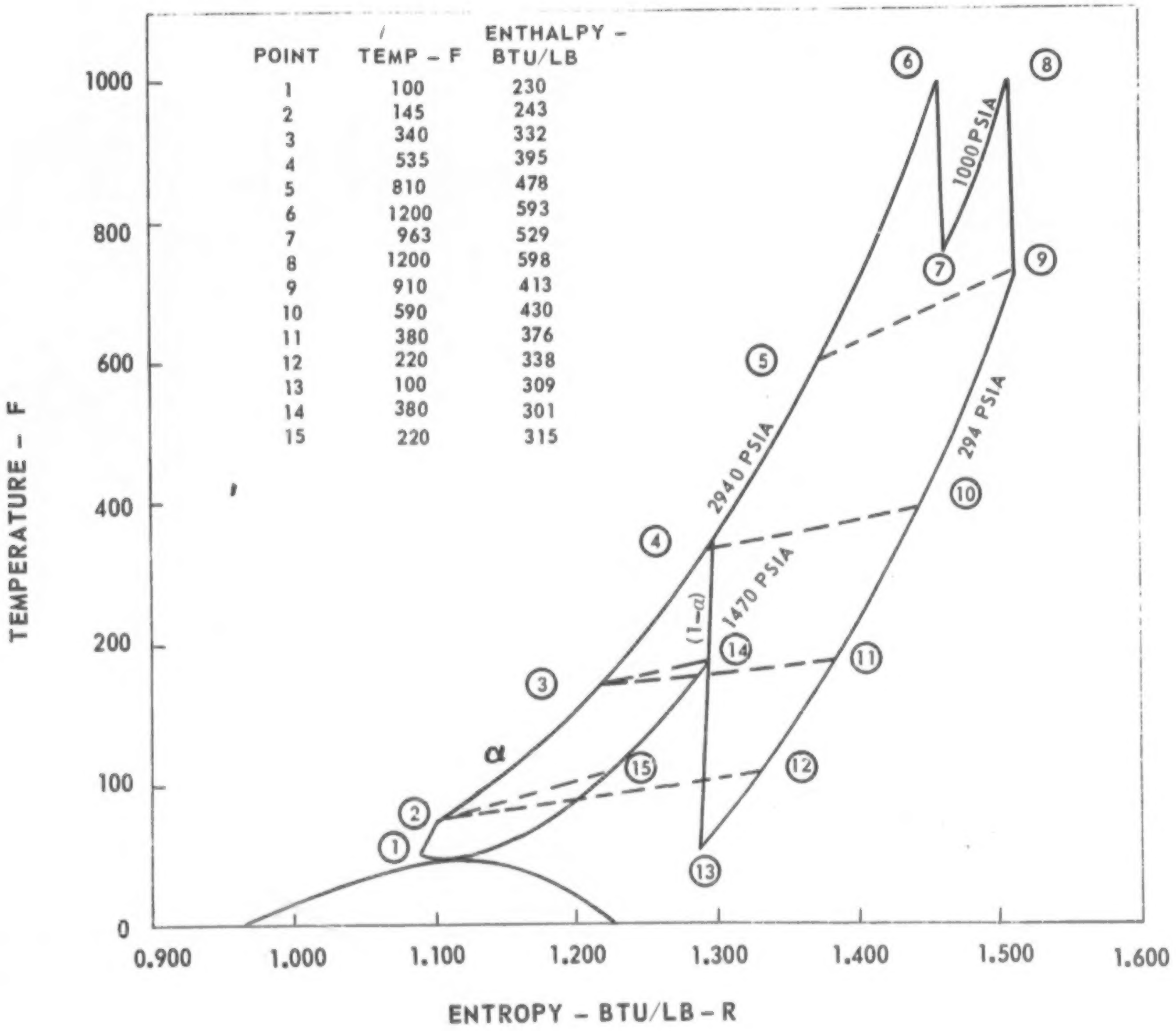
LOW-TEMPERATURE CO_2 CYCLE

$T_{MAX} = 1200$ F

$\xi_r = 85\%$

$\frac{\Delta P}{P} = 15\%$

$\eta_{th} = 44.6\%$



POINT	TEMP – F	ENTHALPY – BTU/LB
1	100	230
2	145	243
3	340	332
4	535	395
5	810	478
6	1200	593
7	963	529
8	1200	598
9	910	413
10	590	430
11	380	376
12	220	338
13	100	309
14	380	301
15	220	315

FIG. 122

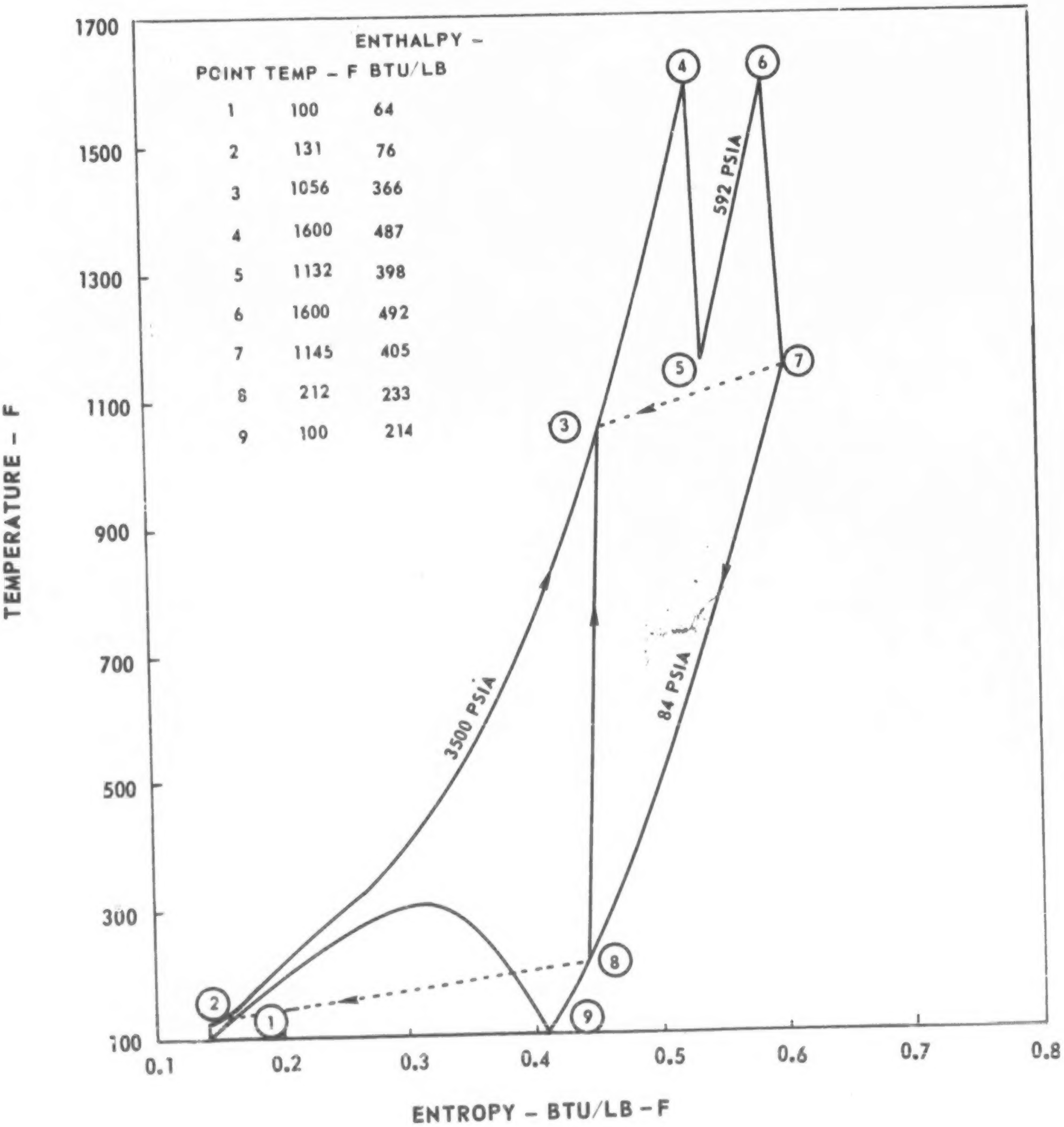
PARTIALLY CONDENSING SO$_2$ CYCLE WITH REHEAT AND REGENERATION

$T_{MAX} = 1600$ F

$\xi_r = 92\%$

$\frac{\Delta P}{P} = 15\%$

$\eta_{th} = 57.7\%$



POINT	TEMP – F	ENTHALPY – BTU/LB
1	100	64
2	131	76
3	1056	366
4	1600	487
5	1132	398
6	1600	492
7	1145	405
8	212	233
9	100	214

FIG. 123

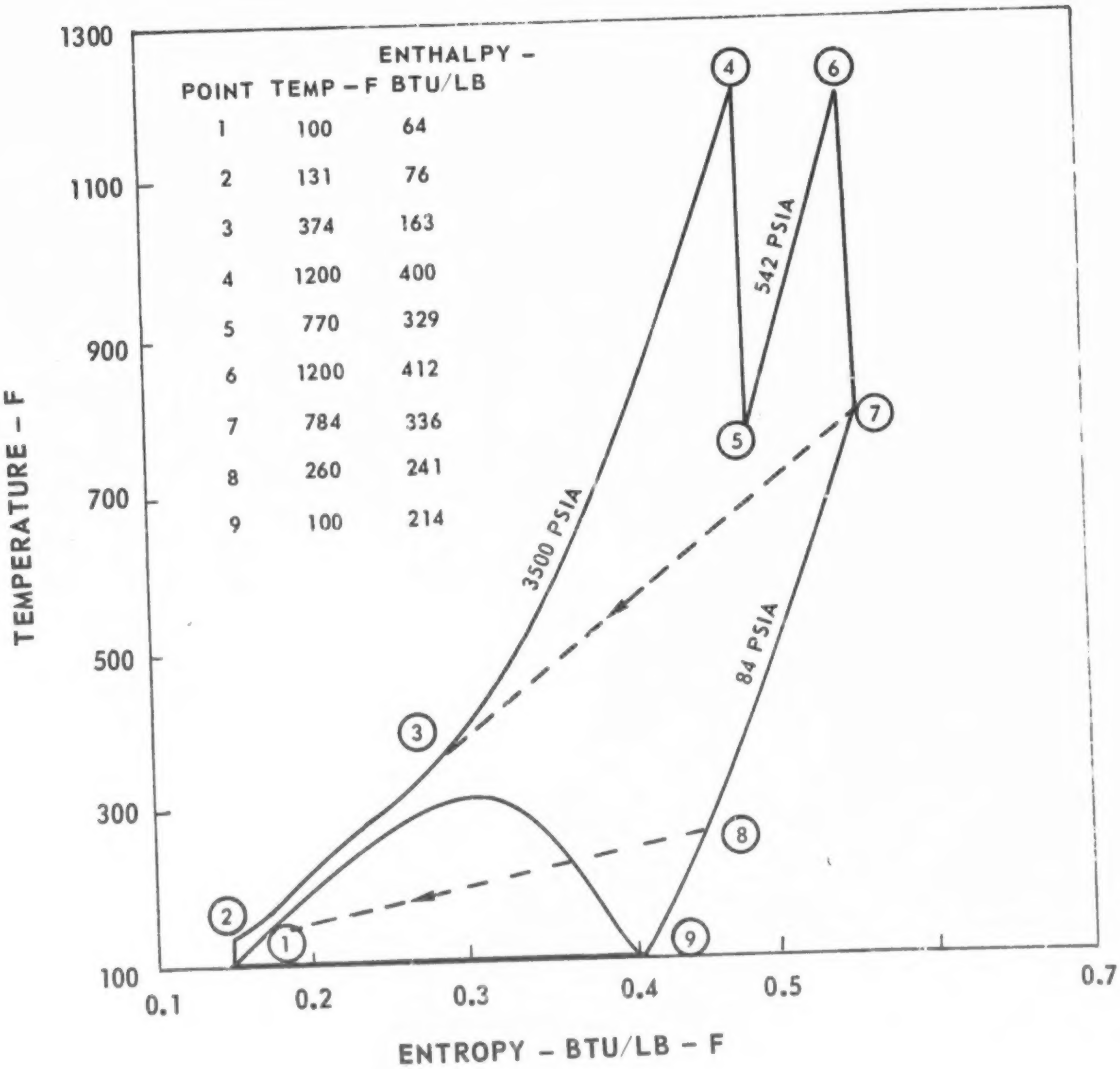
FULLY CONDENSING SO2 CYCLE WITH REHEAT AND REGENERATION

T_{MAX} = 1200 F

ξ_T = 80%

$\frac{\Delta P}{P}$ = 15%

η_{th} = 42.5%

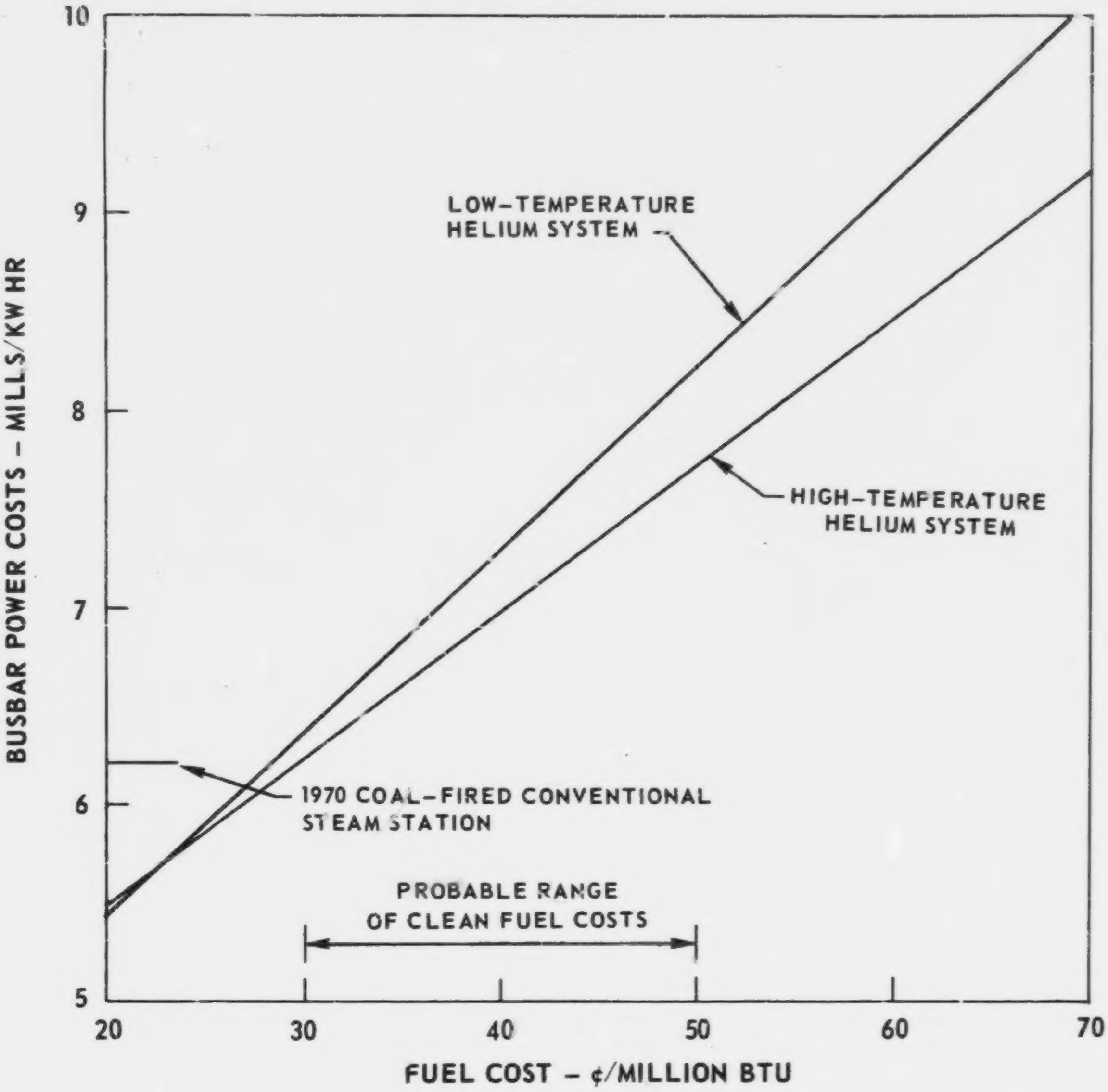


POINT	TEMP – F	ENTHALPY – BTU/LB
1	100	64
2	131	76
3	374	163
4	1200	400
5	770	329
6	1200	412
7	784	336
8	260	241
9	100	214

FIG. 124

COMPARISON OF POWER COSTS FOR HELIUM POWER SYSTEMS

1000-MW NOMINAL SIZE



COMPARISON OF POWER COSTS FOR CARBON DIOXIDE POWER SYSTEMS

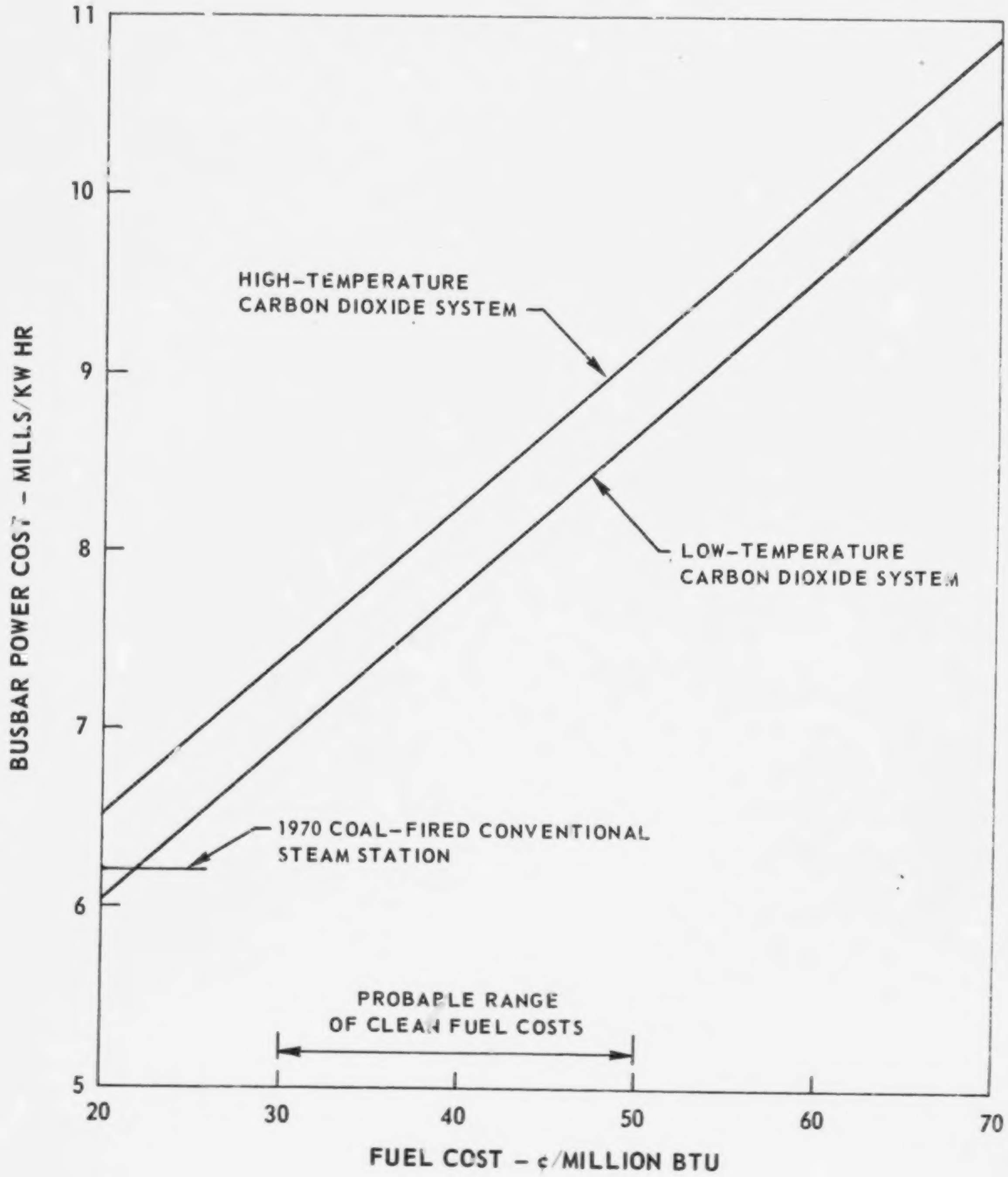
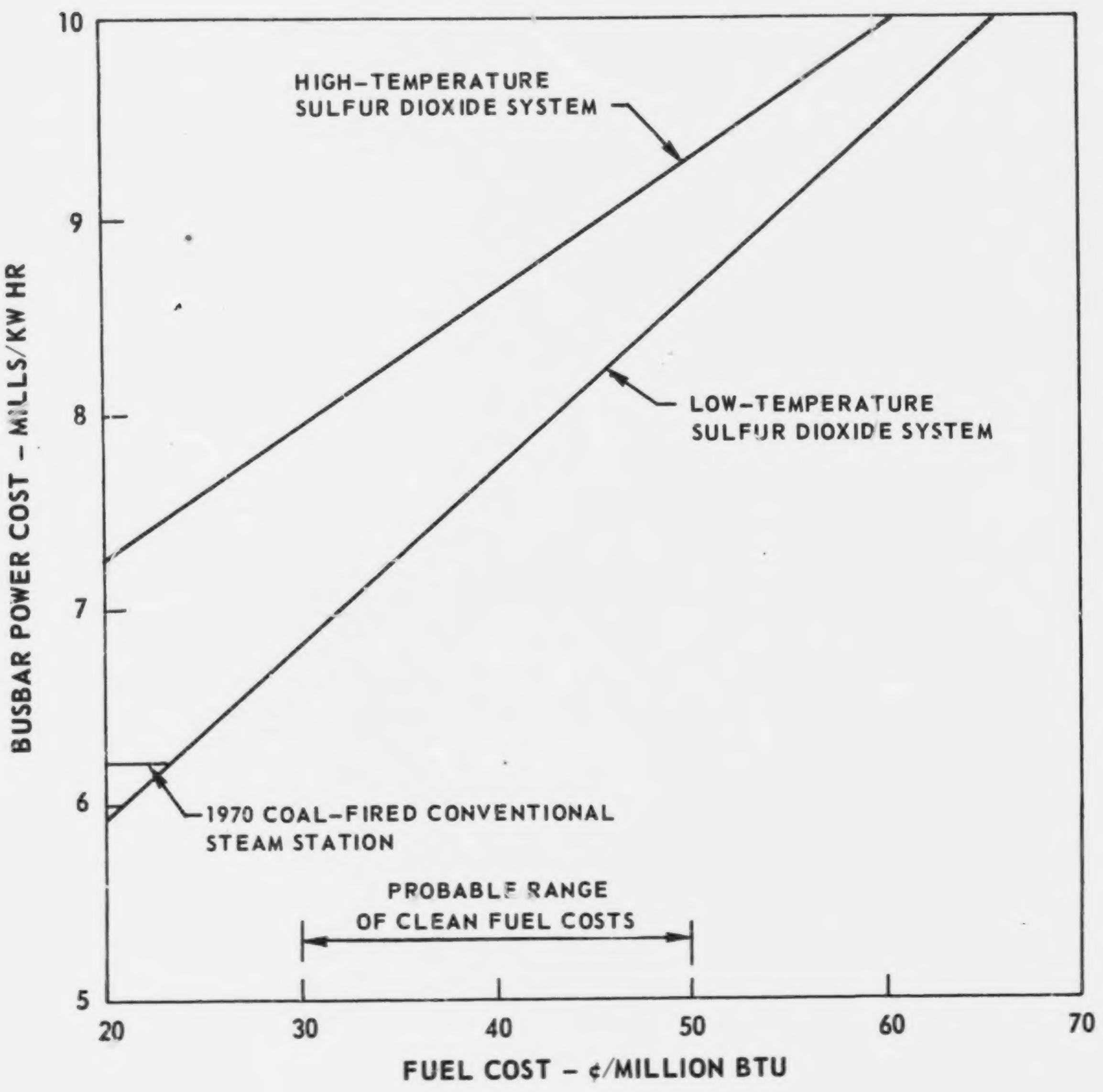


FIG. 126

COMPARISON OF POWER COSTS FOR SULFUR DIOXIDE POWER SYSTEM

1000-MW NOMINAL SIZE



PART THREE - INTEGRATED POWER STATIONS

SUMMARY

An investigation was undertaken to identify those fuel processing systems and advanced-cycle power systems which, in combination, would result in the reduction of sulfur oxide emissions while producing the lowest-cost electric power. The fuel processing systems identified in Part One and the advanced-cycle power systems identified in Part Two as being the most promising were combined, and their integrated performance and economics were determined. Detailed layout drawings of the integrated power station were made, and overall plot plans, including fuel processing systems, were developed. The estimated cost of electricity at the busbar of the power station was determined, and comparisons were made with the projected costs of power from conventional steam power stations burning untreated fuels.

The integrated power stations which appeared to be most promising and which were considered in detail are: for the first generation, COGAS and base-load gas turbine power systems fueled by a Lurgi-type coal gasification system with a hot-potassium carbonate scrubbing system; for the second generation, COGAS and base-load gas turbine power systems fueled by Texaco-type coal and residual oil gasification systems with a potassium carbonate scrubbing system for the coal and a Sulfinol scrubbing system for the oil; and for the third generation, COGAS and base-load gas turbine power systems utilizing the same processes as the second-generation system but with different operating conditions. There is no first-generation residual-oil-fired, advanced-cycle power system because the hydrotreated oil would not be a suitable fuel for use in any high-temperature power system.

SYNOPSIS OF INTEGRATED POWER STATION ANALYSIS RESULTS

An advanced-cycle power station using a COGAS power system fueled by a Texaco-type, entrained-flow, high-temperature gasification system has the potential of generating electric power at busbar costs significantly below those projected for conventional steam power stations while reducing sulfur oxide emissions by over 99%. First-generation, advanced-cycle power stations using a COGAS power system would not be competitive in busbar power costs with conventional steam power stations but would offer approximately 85% reduction in sulfur oxide emissions.

Advanced-cycle power stations using base-load gas turbines fueled by Texaco-type, entrained-flow, high-temperature gasification systems would also be capable of producing electric power at busbar costs below those projected for conventional steam power stations. While the cost of power would be somewhat higher than that projected for COGAS systems, significantly less capital investment would be required for gas turbine stations. Consequently, for operation at reduced load factors (below approximately 50% for second- and third-generation), gas turbine power stations would have the potential of generation the lowest-cost power.

All advanced-cycle gas turbine and COGAS power stations considered would incorporate cooling towers, where needed, in both the fuel processing and power systems; thus, these stations would emit essentially no thermal pollution.

DETAILED INTEGRATED POWER STATION STUDIES

Section I - COGAS Power Stations

This section contains detailed layout, cross-section, and plot-plan drawings for three generations of integrated COGAS power stations. In addition, estimates for the busbar cost of electric power are presented and compared with the projected costs of power generated by conventional steam power stations.

For the most part, the fuel processing system and the COGAS power system could be kept separate since integration of heat exchangers, compressed air sources, or other similar functions of these two parts of the power station would result in no significant cost or performance advantage, but could result in complex piping and manifolding problems. In most fuel gasification processes considered, it would be possible to recover enough waste heat from the fuel process to generate all steam consumed during the fuel conversion. The air compressors for the fuel conversion process were considered to be driven by electric motors with power provided by the power system. One high-temperature fuel conversion process considered for third-generation power systems would require more steam than could be generated by heat recovery in the fuel conversion process, so that a separate section in the waste-heat boiler of the COGAS system was designed to provide this additional steam for the fuel conversion process.

FIRST-GENERATION COGAS STATIONS

Station Design

The optimum fuel for a first-generation COGAS power system is designated Fuel Gas II and would be produced by a Lurgi-type nonentrainment gasification process. This process is fully described in Part One of this report, and the fuel composition and the properties of the products of combustion are presented in Appendix F. The fuel gas would be provided to the gas turbine at a pressure of 21 atm and a temperature of 232 F. Since the pressure ratio for the first-generation COGAS power system would be only 8, it would be desirable to utilize a fuel-expansion turbine to expand the fuel gases from 21 atm to 8 atm and thereby provide some of the power required to drive the fuel process air compressor. However, as described in the subsequent discussion of economics, the first-generation COGAS system would not be economically competitive with conventional coal-fired stations so that this refinement was not included in the final station design.

The power system would consist of three separate gas turbines having a combined power output of 580 Mw and a single steam system with a net output of 431 Mw. Approximately 11 Mw of power system auxiliary power would be required, thereby yielding a net power system output of 1000 Mw. The fuel conversion system would require 83 Mw so that the net integrated power station output would be approximately 918 Mw.

Specific design details for first-generation gas turbines with 2200-F turbine inlet temperature and 8-to-1 pressure ratio are presented in Table 77. These gas turbines would be single-spool engines operating with 40 in. water back pressure. The power turbine would rotate at 1800 rpm and drive an 1800-rpm electric generator. The temperatures of the turbine cooling air, compressor exit air, and exhaust gas are also indicated in this table along with the compressor airflow and exhaust gas flow rates. As shown in this table, the bleed flow required for turbine cooling would be 4.71% of the compressor airflow. The maximum diameter of the gas turbine would occur at the power turbine exit, and this diameter would be slightly over 11 ft. The overall engine length would be slightly over 33 ft. These engines, as well as those for second- and third-generation COGAS stations, could be shipped assembled, except for the inlet casing and exhaust turning section, on a single rail car.

Design details for the first-generation steam system are shown in Table 78. The temperature of the hot gas entering the steam boiler would be approximately

1297 F, and the final stack temperature would be approximately 314 F. Other significant temperatures throughout the boiler are also indicated in this table. The steam turbine throttle flow rate would be slightly over 2.4 million lb/hr at 2400 psig and 1000 F. The heat transfer duties required for each section of the boiler are also presented in Table 78 along with calculated values for the heat transfer surface area in each section.

Figures 127 and 128 depict detailed operating floor plans and cross sections, respectively, of the powerplant building for the first-generation COGAS power systems. It may be seen from these drawings that the power plant building would be very compact, having overall dimensions of only 336 ft x 142 ft. The three gas turbines would be located on the mezzanine floor. All three gas turbines would exhaust into a manifold which would carry the gas turbine exhaust gases to the waste-heat boiler. This boiler would be of outdoor construction and would be located immediately adjacent to the power plant building. After passage of the hot gases through the waste-heat boiler, the gases would be exhausted through a 150-ft stack. Steam generated in the waste-heat boiler would be piped to the tandem-compound, double-flow steam turbine located on the main operating floor.

Considerable detail has been given to auxiliary equipment requirements for the gas turbine, such as the air intake filter and silencing equipment, and to providing sufficient parts laydown area for repair work which might eventually have to be done on these engines. In addition, water-cooled heat exchangers would be installed at grade level, in the space provided beneath the gas turbines, to provide the precooling required for the turbine cooling air. Consideration has also been given to location of the railroad tracks, administration offices, lockers, showers, washrooms, and lavatory. Two separate cranes have been provided, one for the gas turbine equipment and one for the electric generators and steam turbine. In the event that repair work would be required on a gas turbine, it would be lifted and moved to the parts laydown bay.

Figure 129 is a plot-plan drawing for the first-generation power station-utilizing the Lurgi-type gasification of coal. From this drawing it may be seen that the land required for the power plant building comprises only a small fraction of the total land requirement for the power station. Land required for the electrical switchyard, fuel conversion plant, and coal storage facilities would exceed that required by the power plant building. Also shown in this plot plan is the storage tank for a one-week supply of clean, standby oil, which would be required in the event the fuel conversion plant were temporarily shutdown, and the hyperbolic cooling towers, which were included to eliminate thermal pollution from these power stations.

Station Performance

The cycle efficiency of the gas turbine portion of the first-generation power system described in Table 77 would be approximately 27%. The heat recovery steam cycle efficiency, with 132-F feedwater temperature, would be approximately 39%. The resulting overall COGAS power system efficiency (including stack losses, generator inefficiencies and auxiliary loads), therefore, would be 47.0%, as noted in Table 79. Approximately 144.5 million Btu/min of coal energy would be required to supply the fuel conversion process to produce the required amount of Fuel Gas II. Also, approximately 82 Mw of compressor drive power would be required. The net thermal efficiency of the fuel conversion process, including the heat equivalent of the power required to drive the compressor, would then be approximately 76.8%. The net power output from the integrated power station would be approximately 918 Mw, yielding a net overall integrated power station efficiency of approximately 36.1%. This is approximately ½ percentage point less than the efficiency of the first-generation coal-fired steam station described in Part Two, Section I.

Station Economics

The method of determining power system cost is described in Appendix D and was used as a basis for the costs given in Table 101. Likewise, the cost of the fuel processing plant was determined as outlined in Appendix D. COGAS power stations could be constructed in significantly less time than conventional steam stations, 36 months compared to approximately 48 months. The busbar cost of power, which is equivalent to the total annual owning and operating costs of the power station, was found by determining the capital charges on the total system capital, e.g., the installed cost of the power system plus the fuel processing system, and summing these with the operating costs. The operating costs include projected costs for supplies, maintenance, and operation of the fuel processing and power systems plus the cost of the raw fuel, in this case coal at 20¢/million Btu. These costs are summarized in Table 102 along with the busbar power costs for first-generation steam systems and first-generation base-load gas turbine systems.

The projected busbar cost of approximately 7.32 mills/kwhr is about 1.1 mills/kwhr (18%) higher than the projected cost for the conventional steam station burning untreated fuel. This result is shown in graphical form in Fig. 130. Also shown in Fig. 130 is the projected busbar cost for steam power stations having stack gas clean-up devices. A projected incremental cost of 0.3 mills/kwhr and 85% sulfur oxide removal was assumed for these latter stations.

SECOND-GENERATION COGAS STATIONS

Station Design

The fuel for second-generation COGAS power stations was designated Fuel Gas III. This fuel would be derived by partial oxidation of coal in a slagging, entrainment-type gasifier. A detailed description of the Fuel Gas III fuel conversion process is presented in Part One of this report, and the properties of this fuel and the products of combustion derived from this fuel are summarized in Appendix F. This fuel would be provided to the gas turbine at a pressure of 14 atm and a temperature of 232 F. The slight pressure differential between the fuel supply pressure and the required 12 atm delivered pressure would be required for proper operation of the fuel controls.

The second-generation power systems would consist of two gas turbines, having a combined output of approximately 628 Mw, and a single tandem-compound, double-flow steam system, having a nominal output of 381 Mw. Approximately 9 Mw of power system auxiliary power would be required, thereby leaving a net power system output of 1000 Mw. The fuel conversion system would require approximately 55 Mw of electrical power to drive the fuel conversion plant compressors, thereby leaving a net second-generation power station output of 945 Mw.

Specific design details for the second-generation gas turbines are presented in Table 77. These turbines would operate at 2800 F turbine inlet temperature with a pressure ratio of 12. The exhaust gas temperature in this case would be approximately 1514 F. The bleed flow required for turbine cooling would be almost doubled, to 8.5%. The maximum diameter for this gas turbine would be approximately 11 ft, and the overall length from compressor inlet stage to power turbine exit stage would be approximately 27 ft. Although the power output pwer second-generation engine would be significantly greater than the power outputs from the first-generation engines, the air and gas flow requirements and physical dimensions of the second-generation engines would be significantly less than the corresponding values for first-generation engines. These differences in air and gas flow rates and physical dimensions are a direct result of the very high turbine inlet temperature for the second-generation engine.

The characteristics of the steam system for the second-generation power station are presented in Table 78. The power output and steam flow rates for the second-generation steam system would be slightly less than the corresponding values for the first-generation steam system. However, because of the much higher gas temperature going into the second-generation steam boiler, the heat transfer duties for each section of the second-generation boiler would be considerably higher than the corresponding heat transfer duties for the first-generation boiler. The higher gas-to-steam temperature differential for the second-generation case, however, would allow a slight reduction in the surface areas for each section of the second-generation boiler. The final stack gas temperature in this case would be approximately 219 F.

Figures 131 and 132 present plans and cross sections of the power plant building for the second-generation COGAS power system. The overall dimensions of the power plant building in this case would be only 300 ft x 142 ft. The exhaust gases from the two gas turbines would again be manifolded together and passed through the outdoor waste-heat boiler to the stack. As before, steam generated in the waste-heat boiler would be piped to a tandem-compound, double-flow steam turbine.

Figure 133 presents a plot-plan layout for this second-generation coal-fired power station. As for the first-generation case, land required for the building would be nominal. This drawing also illustrated the four parallel trains required for the fuel processing system. Figure 134 presents a detailed plot plan for a second-generation plant which would utilize partial oxidation of residual oil. In this case, oil would be transported to the power station site by barge or tanker, unloaded, and stored in two large storage tanks. As for the other COGAS systems, a one-week supply of clean, standby oil is provided in the event the fuel conversion plant has to be temporarily shut down.

Station Performance

The cycle efficiency of the second-generation gas turbine, also described in Table 79, is approximately 34%. The heat recovery second-generation steam cycle efficiency, with 145-F feedwater temperature, would be approximately 40%. The overall second-generation COGAS power system efficiency (including stack losses, generator inefficiencies, and auxiliary loads), therefore, would be approximately 54.5%.

The second-generation coal conversion process would require approximately 133 million Btu/min of coal energy, and consume approximately 55 Mw of electric power for compressor drive. The net thermal efficiency of the fuel processing plant, thus, would be approximately 87.4%. The net power station output would be approximately 945 Mw and the overall power station efficiency would be 47.6%. This efficiency is nearly 23% (9 percentage points) higher than the projected efficiency for conventional coal-fired steam power stations.

The second-generation oil conversion process would require approximately 109.9 million Btu/min of oil energy. The oil conversion process design would be slightly different from that for the coal conversion process in that, instead of providing electricity to drive the air compressors, the oil conversion process would generate steam from waste heat recovery to drive the compressors. There would be a slight deficiency, however, in the steam generation rate, and the difference would be made up by steam generated in the COGAS waste-heat steam system boiler. The electric power equivalent of the steam transferred from the power system to the fuel conversion system would be approximately 13 Mw. The overall thermal efficiency of the oil conversion process, including the energy in the steam transferred from the power system, would be approximately 93.8%. The net power station output would be approximately 987 Mw, and the net station efficiency would be approximately 51.1%.

Station Economics

The busbar costs of power from second-generation coal- and oil-fired COGAS stations are given in Table 103 and 104, respectively. The capital costs for the power system and the fuel processing systems are based upon the estimates given in Table 88 and Table 25. The projected busbar power costs presented in Tables 103 and 104 and in Figs. 130 and 135 for coal- and oil-fired COGAS power stations are approximately 9% and 13% less than corresponding costs for conventional steam power stations.

THIRD-GENERATION COGAS STATIONS

Station Design

As explained in Part One of this report, the high-temperature fuel intended for third-generation power stations, Fuel Gas IV, results in higher fuel costs than Fuel Gas III which was selected for second-generation power stations. Consequently, Fuel Gas III was also selected for third-generation power stations.

The third-generation power station would also utilize two gas turbines having a combined output of 696 Mw. The steam system would have a nominal output of 312 Mw. Approximately 8 Mw of power system auxiliary power would be required, thereby yielding a net power system output of 1000 Mw. The fuel conversion plant would consume approximately 59 Mw, thereby yielding a net integrated power station output of 941 Mw.

Detailed design characteristics of the third-generation gas turbine are presented in Table 77. These engines would be twin-spool engines with a pressure ratio of 20 and a turbine inlet temperature of 3100 F. Because of the high pressure ratio of these engines, the exhaust gas temperature would be slightly less than that for second-generation engines, or 1485 F. The compressor airflow and exhaust gas flow rates would also be less than those for second-generation engines. The bleed flow required for turbine cooling would be approximately 9%. The physical dimensions of the third-generation engine would be slightly less than the coresponding physical dimensions for second-generation engines.

The characteristics of the third-generation steam systems are presented in Table 78. Because of the reduced steam system output, the throttle and reheat steam flow rates would be reduced to approximately 1,750,000 lb/hr and 1,728,000 lb/hr, respectively. These reduced gas temperature and steam flow rates would result in a reduction of the heat transfer duty for each section of the boiler below the corresponding values for second-generation boilers. The stack gas temperature for the third-generation case would be approximately 241 F. This stack temperature is higher than for the second-generation case for two reasons. First, the high pressure ratio of these engines would result in lower boiler gas inlet temperature which in turn would lead to the higher stack temperatuer as indicated by the results in Fig. 77. Secondly, because of the higher pressure ratio, the compressor exit temperature would be increased to approximately 861 F. Since more heat would have to be removed from this compressor air to achieve the 200-F precooling air temperature required, the temperature of the feedwater entering the boiler would be increased to approximately 159 F. These two effects would combine to produce the higher stack temperature. If a lower third-generation pressure ratio (16-to-1) had been utilized, the stack temperature could have been maintained in the 200- to 220-F range.

Since the physical dimensions of the third-generation gas turbines, steam turbine, and boiler equipment would change very little from the corresponding dimensions for the second-generation equipment, the third-generation power station building layout, cross sections, and power station plot plan would not change significantly from the corresponding drawings for second-generation power plants. Consequently, Figs. 131 through 134 correspond to both second-generation and third-generation power plants. Also included for comparative purposes is a plot for a coal-fired station with a dolomite process desulfurization system. This is shown in Fig. 136.

Station Performance

The cycle efficiency of the gas turbine portion of the third-generation power system, described in Table 77, would be approximately 40.4%. The heat recovery steam cycle efficiency with 159-F feedwater temperature would be approximately 40.1%. The overall COGAS power system efficiency (including stack losses, generator inefficiencies, and auxiliary loads), therefore, would be approximately 57.7%, as noted in Table 79.

The third-generation coal conversion process would require approximately 106.4 million Btu/min of coal energy and consume approximately 67 Mw of compressor power. Although this third-generation coal conversion process would be essentially the same as the second-generation coal conversion process, approximately 12 Mw of additional compressor power would be required in this case because of the higher operating pressure level of the third-generation power system. The overall thermal efficiency of the fuel conversion process would be approximately 86.6%. The net station output in this case would be approximately 930 Mw, and the overall power station efficiency would be approximately 49.9%. This efficiency is approximately 26% (10.3 percentage points) higher than the efficiency projected for the third-generation conventional, coal-fired steam power station.

The third-generation oil conversion process would require approximately 103.8 million Btu/min of oil energy and require the equivalent of approximately 22 Mw of steam from the waste-heat steam system. The net thermal efficiency of this fuel conversion process would be approximately 92.9%. The net power output from the third-generation oil-fired station would be approximately 978 Mw and the overall power station efficiency would be approximately 53.6%.

Station Economics

The projected busbar costs for the third-generation COGAS system are given in Table 105 and 106 for coal-fired and oil-fired stations, respectively. The coal-fired station busbar cost of 5.14 mills/kwhr is approximately 10% less than that of the projected conventional coal-fired station. The oil-fired station busbar cost is about 15% lower than that for the conventional oil-fired station. These are shown graphically in Figs. 130 and 135 for coal- and oil-fired stations, respectively.

Section II - Base-Load Gas Turbine Stations

Section II contains the detailed power plant layout, cross sections, and overall plot plans for three generations of integrated, base-load gas turbine power stations. The busbar costs of electric power are developed and compared with the costs of power generated by conventional steam power stations.

As in the case of the COGAS stations (Part Three, Section I), the fuel conversion and power systems could be separated. Since there would be no steam generated by the power station, interchange of steam between the fuel processing and power systems would not be possible.

FIRST-GENERATION STATIONS

Station Design

The optimum fuel for the first-generation base-load gas turbine power plant is designated Fuel Gas II and would be produced by a Lurgi-type nonentrainment gasification system. The fuel gas would be supplied to the turbine at 21 atm and a temperature of 232 F.

The power system would consist of five gas turbines, each driving a 200-Mw (nominal) generator. A total of approximately 109 Mw of auxiliary power and power for the fuel processing system would be required; thus the net station output would be about 891 Mw.

Specific design details of the first-generation gas turbines with 2200 F turbine inlet temperature and 20-to-1 pressure ratio are given in Table 60. The gas turbines would be twin-spool engines with the generator drive turbines rotating at 1800 rpm. As shown in Table 60, the bleed flow required for turbine cooling would be 5.72% of the compressor airflow. The maximum diameter would be slightly over 10 ft at the power turbine, and the overall length would be such that the engine could be shipped, assembled, on a single rail car.

Figure 137 presents a detailed layout drawing for the oeprating floor plan of the first-generation, base-load gas turbine power staton, and a detailed cross-section of this plant is shown in Fig. 138. All five gas turbines would exhaust directly into silencers contained in each of the five stacks. The height to the top of the stack would be approximately 122 ft. From the overall plot plan, illustrated in Fig. 139, it can be seen that the power plant building, which would measure only 268.5 ft x 143 ft, would occupy only a very small portion of the land area. Since all heat rejected by the turbine would be directed to the atmosphere, large cooling towers for the power system would not be needed. Relatively small-capacity cooling towers would be required to reject heat from the fuel conversion plant, however.

Station Performance

The efficiency of the gas turbine for a first-generation power system described in Table 60 would be approximately 35.2%. Approximately 192.9 million Btu/min of coal energy would be required as input to the fuel conversion process. Also, approximately 109 Mw of compressor drive power would be required. The net thermal efficiency of the fuel conversion process, including the heat equivalent of the power required to drive the compressor, would be approximately 74.8%. The net overall station efficiency would then be 26.3%.

As in the case of the COGAS system, the hydrotreated oil would not be a suitable fuel.

Station Economics

The method of determining the busbar cost of power is similar to that described in the discussions of COGAS systems. The gas turbine station would also have the advantage of significantly shorter construction time, 30 months vs 48 for conventional steam stations, which in turn results in lower interest and escalation charges. The projected busbar power cost of 7.98 mills/kwhr is appreciably higher than that of the conventional coal-fired steam station, and about 10% higher than the COGAS system (Table 102). This comparison is shown in Fig. 130.

Load factor has a very large effect on the busbar cost of power. While fuel costs, maintenance, supplies, etc., remain essentially constant in terms of mills/kwhr at all load factors, the fixed costs are a hyberbolic function of load factor. This relationship is shown in Fig. 140, where the busbar costs are shown as a function of load factor for three first-generation power stations--a conventional coal-fired steam station, a COGAS station, and a base-load gas turbine station. From this figure, it can be seen that only at very low load factors, less than approximately 25%, would the gas-turbine stations have lower power costs than either the COGAS or conventional coal-fired steam stations. It must be noted, however, that the power stations considered herein were designed for base-load operation and, consequently, would not be the optimum stations for use in low-load-factor applications.

SECOND-GENERATION STATIONS

Station Design

The fuel for the second-generation, base-load gas turbine power station is designated Fuel Gas III and could be produced by a Texaco-type, entrained-flow gasification system. The fuel gas would be supplied to the turbine at 34 atm and 232 F.

The power system would consist of three gas turbines, each driving a nominal 337-Mw generator. A total of approximately 102 Mw of auxiliary power and power for the coal-fired fuel processing system would be required; thus the net station output would be 910 Mw. For an oil-fired station, the net output would be approximately 967 Mw.

Specific design details of the second-generation gas turbines with 2800 F turbine inlet temperature and 32-to-1 pressure ratio are given in Table 60. The gas turbine would be a twin-spool engine with the generator drive turbine rotating at 1800 rpm. The bleed flow for cooling would be 10.8% of the compressor air flow. The maximum diameter would be slightly over 10 ft at the power turbine, and the overall length would be nearly 26 ft.

Figure 141 presents a detailed layout drawing of the operating floor for the second-generation station. A cross section of this plant is shown in Fig. 142, and an overall plot plan for the integrated coal gasification and power system is shown in Fig. 143. A similar drawing for the integrated oil gasification and power station is shown in Fig. 144. The power plant building for the second-generation gas turbine system would be very compact, measuring 196 ft by 140 ft. All three gas turbines would exhaust directly into silencers contained in each of the three stacks.

Station Performance

The efficiency of the gas turbines for the second-generation power system would be approximately 41.5%. Approximately 148 million Btu/min of coal energy would be required as input to the fuel conversion process, as would about 90 Mw of compressor drive power. The net thermal efficiency of the fuel conversion process, including compressor power, would then be approximately 84.3%. The net overall station efficiency would thus be 35%.

Station Economics

The projected busbar power costs for the second-generation base-load gas turbine station are approximately 5.43 mills/kwhr and 4.760 mills/kwhr for the coal- and oil-fired stations, respectively (Tables 103 and 104). These costs are slightly lower than those projected for the conventional coal- and oil-fired steam stations (Fig. 130 for coal-fired and Fig. 135 for oil-fired stations).

Figure 145 shows the effect of load factor. Unlike the first-generation gas-turbine stations, the second-generation stations would demonstrate busbar costs equal to or lower than the costs of both the conventional coal-fired steam station and the COGAS station at the extreme lower end of the probable range of operation. Thus, the gas-turbine station could be a logical choice for stations with load factors below the 50 to 60% range.

THIRD-GENERATION STATIONS

Station Design

The fuel for the third-generation base-load, gas turbine power station would again be Fuel Gas III. This gas would be supplied to the turbine at 36 atm and 232 F.

The power system would consist of two 506-Mw nominal size gas turbines. A total of 112 Mw of auxiliary power and compressor power would be required for the coal-fired fuel conversion system; thus the net station output would be 900 Mw. The oil-fired station would have a net output of 945 Mw.

Specific design details of the third-generation gas turbines with 3100 F turbine inlet temperature and a 34-to-1 pressure ratio are given in Table 60. The engine would be of twin-spool design with a 1800-rpm power turbine. The bleed flow ould be 10.7% of the compressor flow. The maximum diameter, at the power turbine, would be nearly 11 ft, and the overall length approximately 29 ft.

Figure 146 presents a plan view of the operating floor, and a cross section is given in Fig. 147. An overall plot plan for an integrated coal-fired station is given in Fig. 148. This plot plan was prepared for the high-temperature dolomite fuel conversion process in order to show the differences between this and the second-generation station (Fig. 144) which would be identical to the plot plan for the third-generation system with the hot-carbonate scrubbing process. The plot plan for the third-generation, oil-fired station would be identical to that of the second-generation oil-fired system (Fig. 144).

Station Performance

The efficiency of the gas turbine for a third-generation station would be approximately 43.6%. Approximately 144.6 million Btu/min of coal energy would be required as input to the fuel conversion system, as would 100 Mw of electrical power for compressor drive. The net thermal efficiency of the fuel conversion system would be 81.2% and the overall thermal efficiency would be 35.4%.

For the oil-fired station, 55 Mw of compressor drive power and 138 million Btu/min of oil would be required. The net thermal efficiency of the fuel processing system would thus be 89.2% and the net thermal efficiency of the power station would be 38.9%.

Station Economics

The busbar costs of power from third-generation, base-load gas turbine stations are given in Tables 105 and 106 as approximately 5.41 mills/kwhr and 5.73 mills/kwhr for coal-fired and oil-fired stations, respectively. These costs are also shown graphically in Figs. 130 and 135. Again the gas turbine station busbar costs are more attractive than the coal station but slightly more costly than the COGAS within the probable operating range. At lower load factors, as shown in Fig. 149, the gas-turbine station can be more attractive than the COGAS. However, unlike the second-generation stations, the load factor at which the gas-turbine becomes more attractive is below the probable range of operation.

CONCLUDING REMARKS

The objective of the study reported herein was to identify those combinations of fuel cleanup processes and advanced-cycle power systems which would result in the lowest overall cost of producing electric power while reducing or eliminating emissions of sulfur oxide pollutants for fossil-fueled central power stations. The results of this study indicate that prospective power stations based on the use of advanced-cycle gas turbines, either used alone or in combination with waste-heat recovery steam systems, and burning a gasified and desulfurized fuel, have the potential of producing electric power at costs significantly lower than those projected for conventional steam power stations while simultaneously reducing the emissions of sulfur oxides by over 99% compared to present-day emissions. First-generation coal-fired gas turbine and COGAS stations would have approximately 28.4% and 17%, respectively, higher power costs than coal-fired steam stations, but 85% lower sulfur oxide emissions. Second- and third-generation advanced power stations would be capable of reducing sulfur oxide emissions by 98% and 99%, respectively, while producing lower-cost power than could be produced by conventional steam stations. Second- and third-generation gas turbine power stations would have approximately 5% lower power cost than coal-fired steam stations and 9% lower cost than residual-oil-fired steam stations. Second- and third-generation COGAS stations would have even lower power costs, with the corresponding costs being approximately 10% and 15% lower, respectively, than those of coal- and oil-fired steam stations.

Additional advantages of advanced-cycle gas turbine and COGAS power stations involve reduced construction time and lower thermal pollution and emissions of nitrogen oxides. Since gas turbines upon which these advanced station designs are based could be factory-assembled before shipment to the station site, construction times could be reduced to approximately 30 months and 36 months for gas turbine and COGAS stations, respectively, compared to 48 months for steam stations. Also, since cooling towers would be included to reject any waste heat from the fuel processing or power systems which would otherwise be rejected to cooling water, these power stations essentially would not cause any thermal pollution. Furthermore, because of the use of gas turbines and the high thermal efficiencies attainable with COGAS systems, the amount of heat which would have to be rejected by these cooling towers would be considerably less (from approximately 45% to 85%, depending on station configuration and design) than the corresponding heat rejection by steam stations, so that the costs and problems associated with performance degradation, makeup water supply, and public acceptance would be considerably mitigated. The final advantage of these advanced-cycle power stations regards the potential of significantly reducing emissions of nitrogen oxides. Although investigation of these emissions was beyond the scope of the present study, basic theoretical investigations currently in progress at the Research Laboratories (see Appendix J) indicate that it might be possible to design new gas turbine burners which would reduce emissions of nitrogen oxides by a significant amount compared to present-day emissions.

TABLE 102

COST SUMMARY FOR FIRST-GENERATION, COAL-FIRED POWER STATIONS

	Steam(1)	Gas-Turbine	COGAS
Net Power Station Output, Mw	1012	891	918
Capital Investment, $1000			
Power System	176,953	77,514	109,261
Fuel Processing System		51,500	44,500
Total Station	176,953	129,014	153,761
Net Capital Cost, $/kw	174.8	144.8	167.5
Annual Owning and Operating Cost mills/kwhr			
Capital Charges(2)	3.990	3.306	3.824
Operation, Supplies, and Maintenance			
Power System	0.372	0.786	0.623
Fuel Processing System(3)		0.900	0.700
Fuel @ 20 ¢/million Btu	1.865	2.990	2.170
Busbar Power Cost	6.227	7.982	7.317

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(1) Does not include sulfur oxide emission control equipment.
(2) Capital charges @ 14% per annum @ 70% load factor.
(3) Includes credit for sulfur.

TABLE 103

COST SUMMARY FOR SECOND-GENERATION, COAL-FIRED POWER STATIONS

	Steam(1)	Gas-Turbine	COGAS
Net Power Station Output, Mw	1012	910	945
Capital Investment, $1000			
Power System	162,674	60,874	94,005
Fuel Processing System		32,000	26,300
Total Station	162,674	92,874	120,305
Net Capital Cost, $/kw	160.7	102.0	127.3
Annual Owning and Operating Cost, mills/kwhr			
Capital Charges(2)	3.668	2.330	2.906
Operation, Supplies, and Maintenance			
Power System	0.322	0.767	0.626
Fuel Processing System(3)		0.340	0.270
Fuel @ 20¢/million Btu	1.768	1.990	1.440
Busbar Power Cost	5.758	5.427	5.242

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(1) Does not include sulfur oxide emission equipment.
(2) Capital charges @ 14% per annum @ 70% load factor.
(3) Includes credit for sulfur.

TABLE 104

COST SUMMARY FOR SECOND-GENERATION, OIL-FIRED POWER STATIONS

	Steam(1)	Gas-Turbine	COGAS
Net Power Station Output, Mw	1021	967	987
Capital Investment, $1000			
Power System	152,700	60,874	94,005
Fuel Processing System		24,200	19,700
Total Station	152,700	85,074	113,705
Net Capital Cost, $/kw	149.4	88.0	115.2
Annual Owning and Operating Cost, mills/kwhr			
Capital Charges(2)	3.412	1.942	2.596
Operation, Supplies, and Maintenance			
Power System	0.312	0.815	0.654
Fuel Processing System(3)		0.328	0.280
Fuel @ 30¢/million Btu	2.625	2.675	2.000
Busbar Power Cost	6.349	5.760	5.530

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(1) Does not include sulfur oxide emission control equipment.
(2) Capital charges @ 14% per annum @ 70% load factor.
(3) Includes credit for sulfur.

TABLE 105

COST SUMMARY FOR THIRD-GENERATION COAL-FIRED POWER STATIONS

	Steam(1)	Gas-Turbine	COGAS
Net Power Station Output, Mw	1012	900	933
Capital Investment, $1000			
Power System	162,674	61,712	89,269
Fuel Processing System		31,700	26,400
Total Station	162,674	93,412	115,669
Net Capital Cost, $/kw	160.7	103.8	124.0
Annual Owning and Operating Cost, mills/kwhr			
Capital Charges(2)	3.668	2.369	2.830
Operation, Supplies, and Maintenance			
Power System	0.322	0.777	0.634
Fuel Processing System(3)		0.370	0.310
Fuel @ 20¢/million Btu	1.724	1.890	1.370
Busbar Power Cost	5.714	5.406	5.144

---

(1) Does not include sulfur oxide emission control equipment.
(2) Capital charges @ 14% per annum @ 70% load factor.
(3) Includes credit for sulfur.

TABLE 106

COST SUMMARY FOR THIRD-GENERATION OIL-FIRED POWER STATIONS

	Steam(1)	Gas-Turbine	COGAS
Net Power Station Output, Mw	1021	945	978
Capital Investment, $1000			
Power System	152,700	61,712	89,269
Fuel Processing System		23,700	18,900
Total Station	152,700	85,412	108.169
Net Capital Cost, $/kw	149.5	90.4	110.6
Annual Owning and Operaging Cost, mills/kwhr			
Capital Charges(2)	3.412	1.950	2.470
Operation, Supplies, and Maintenance			
Power Systm	0.312	0.796	0.648
Fuel Processing System(3)		0.350	0.300
Fuel @ 30¢/million Btu	2.560	2.630	1.850
Busbar Power Cost	6.284	5.726	5.268

---

(1) Does not include sulfur oxide emission control equipment.
(2) Capital charges @ 14% per annum @ 70% load factor.
(3) Includes credit for sulfur.

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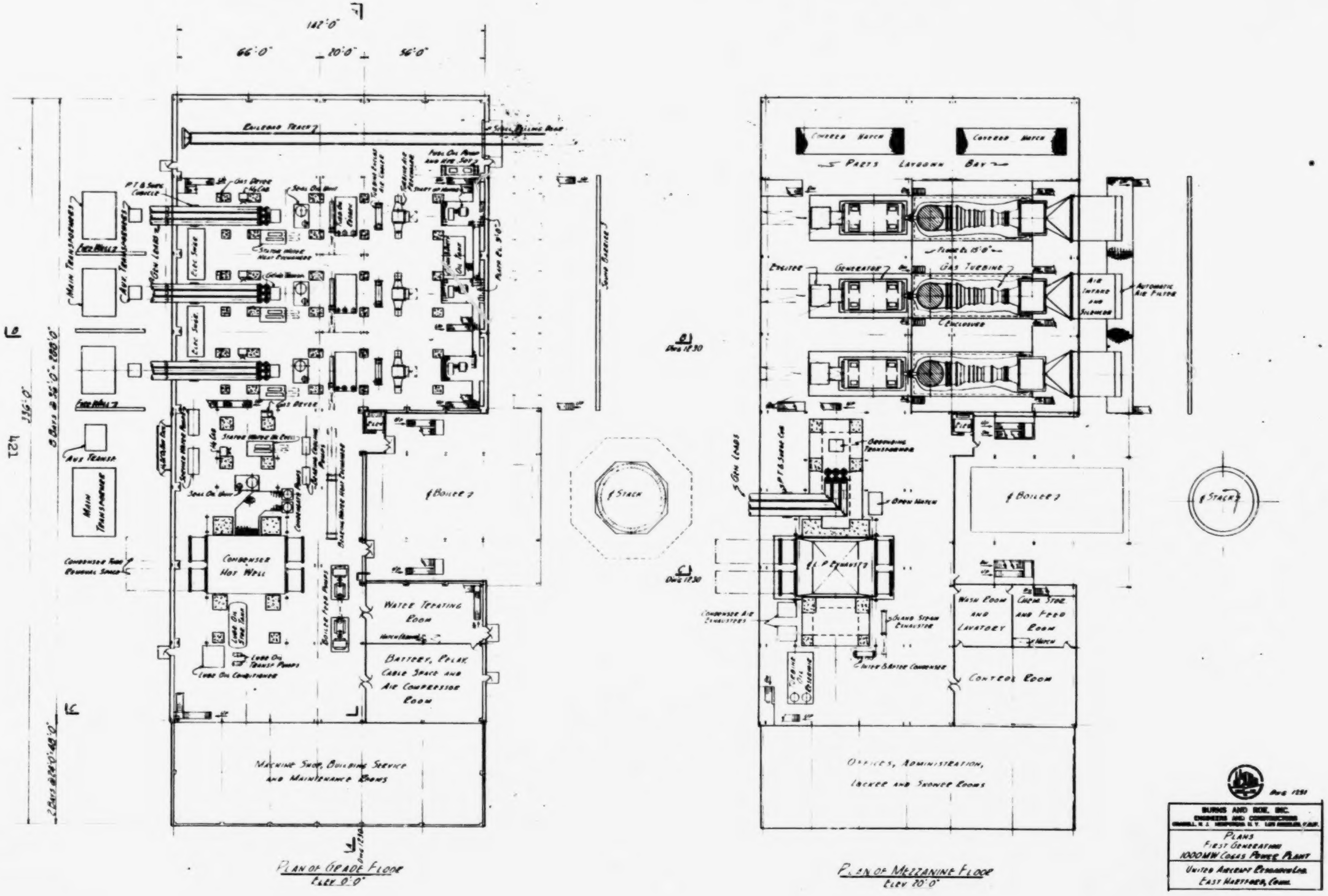

Plan of Grade Floor
Elev 0'-0"

Plan of Mezzanine Floor
Elev 20'-0"

Aug 1951

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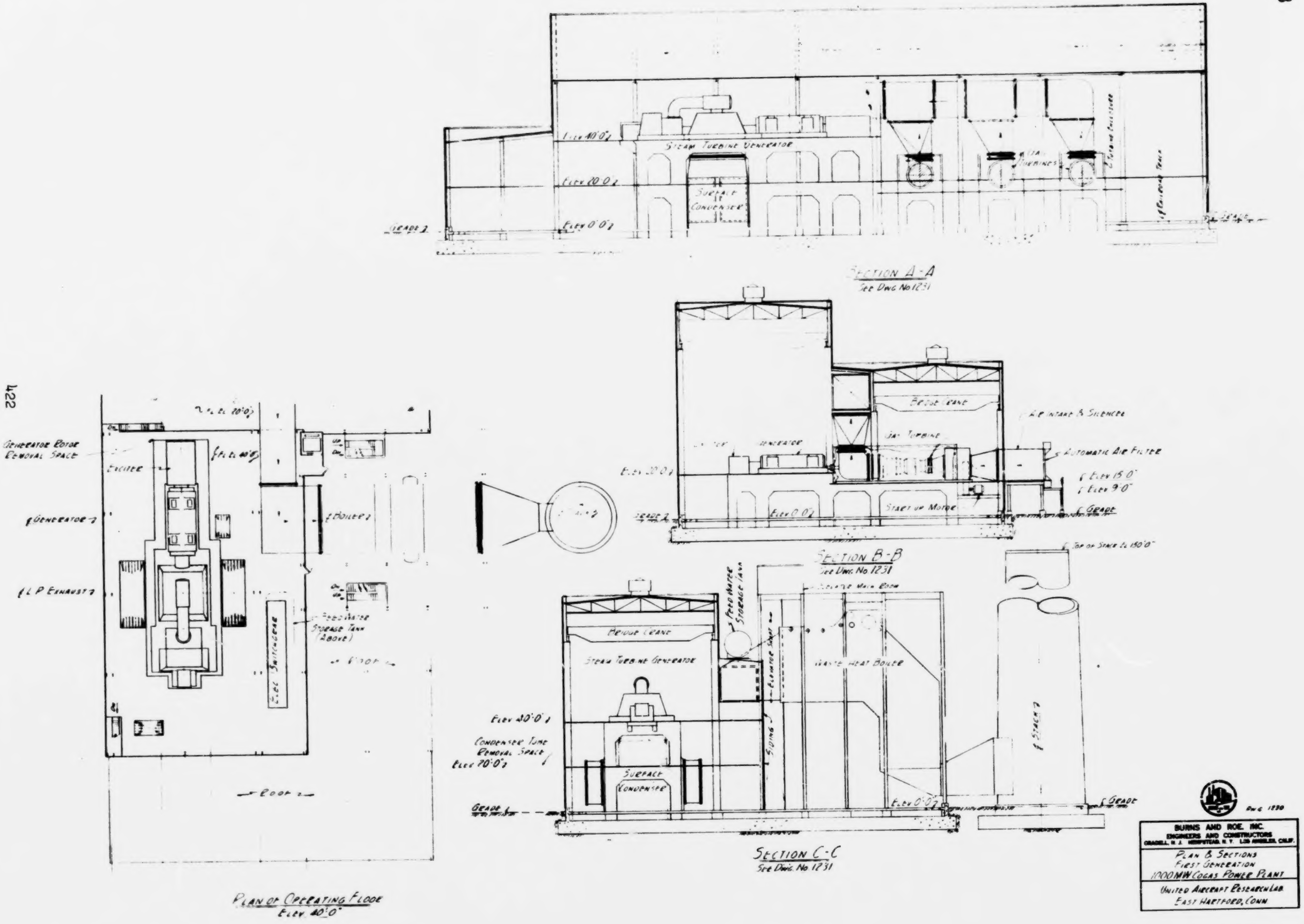
ENGINEERS AND CONSTRUCTORS

Plans
First Generation
1000 MW Cogas Power Plant

United Aircraft Research Lab
East Hartford, Conn.

FIG. 127

FIG. 128



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ORADELL, N. J. HEMPSTEAD, N. Y. LOS ANGELES, CALIF.

Plan & Sections
First Generation
1000MW Cogas Power Plant

United Aircraft Research Lab
East Hartford, Conn

NOT REPRODUCIBLE

FIG. 129

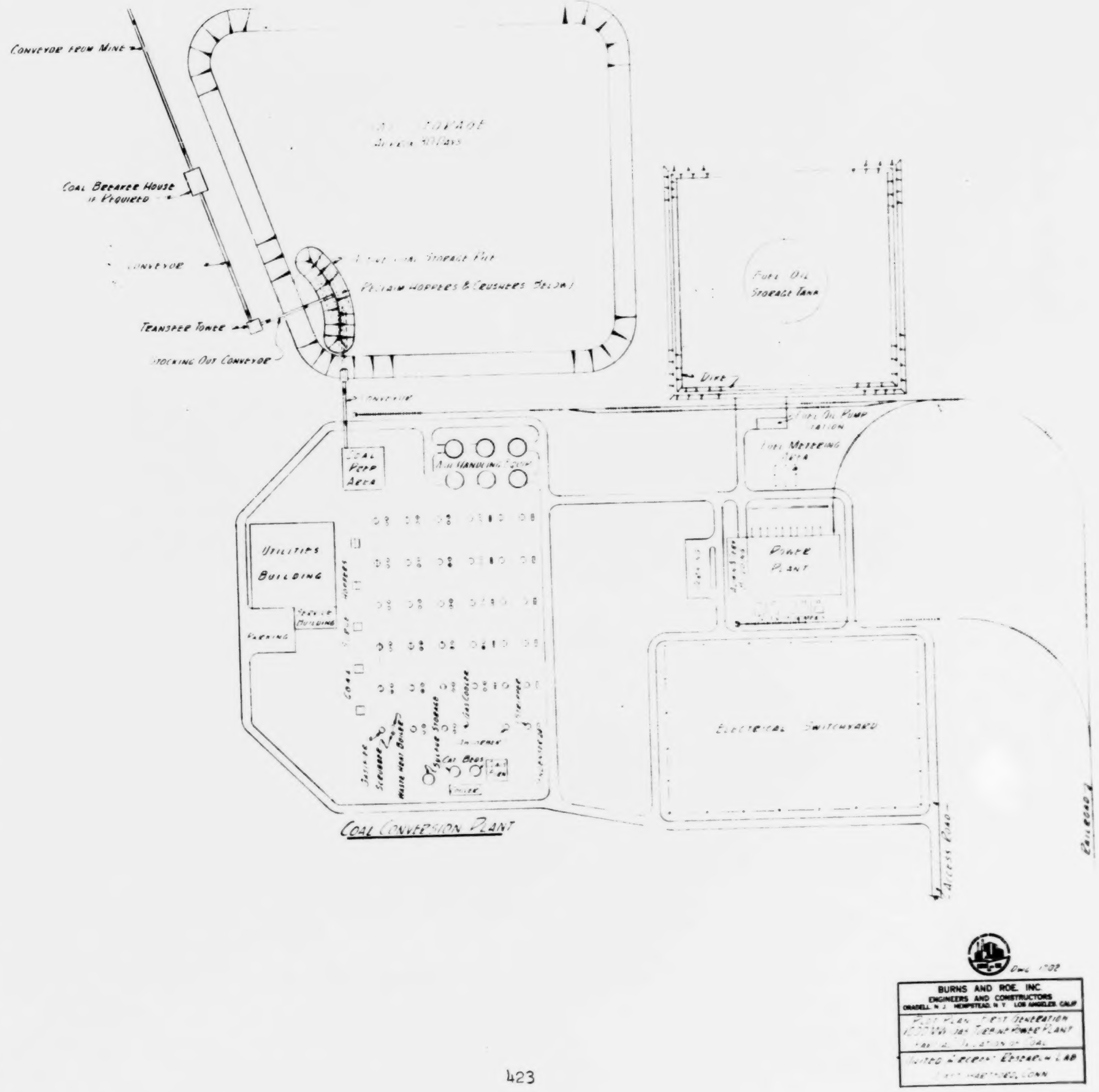
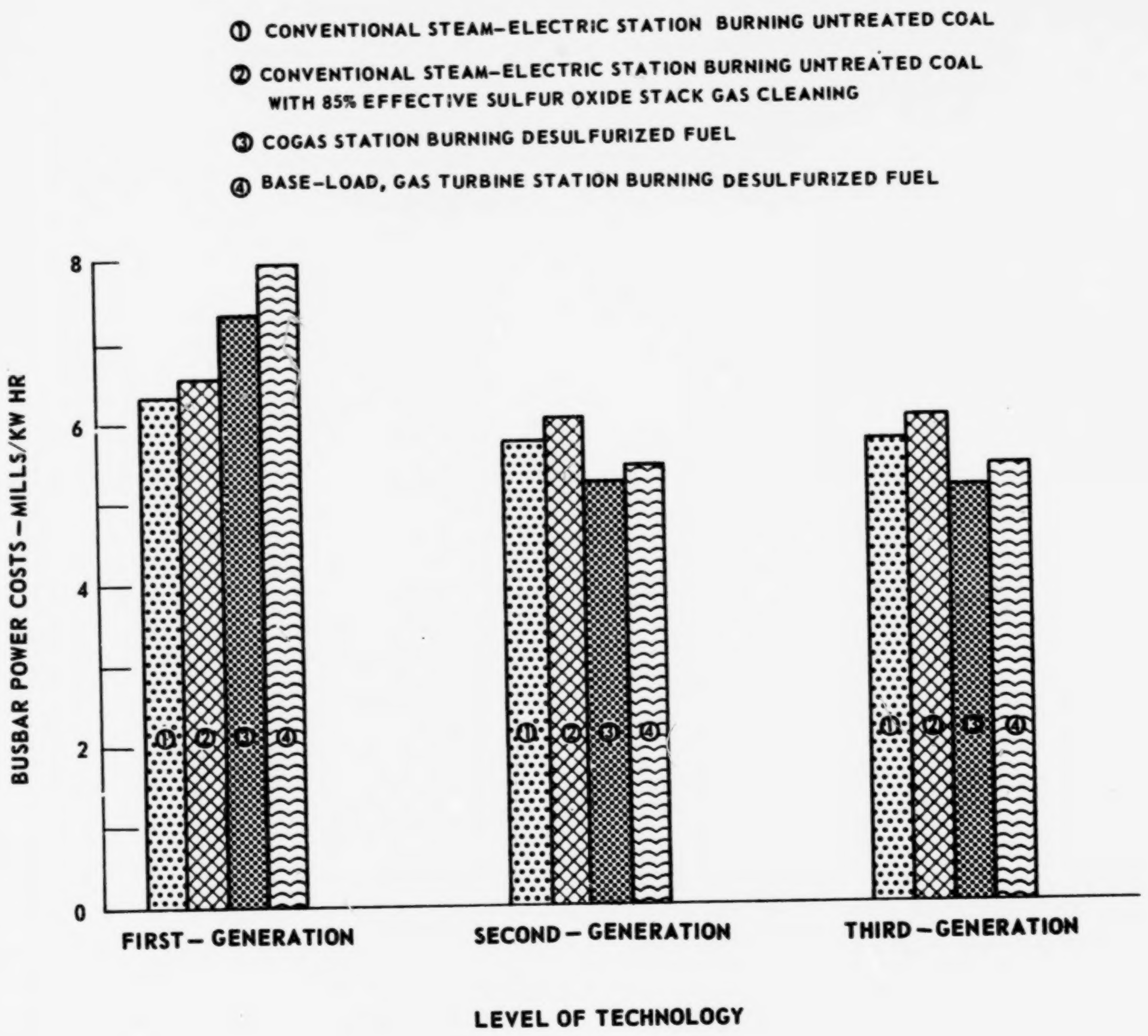
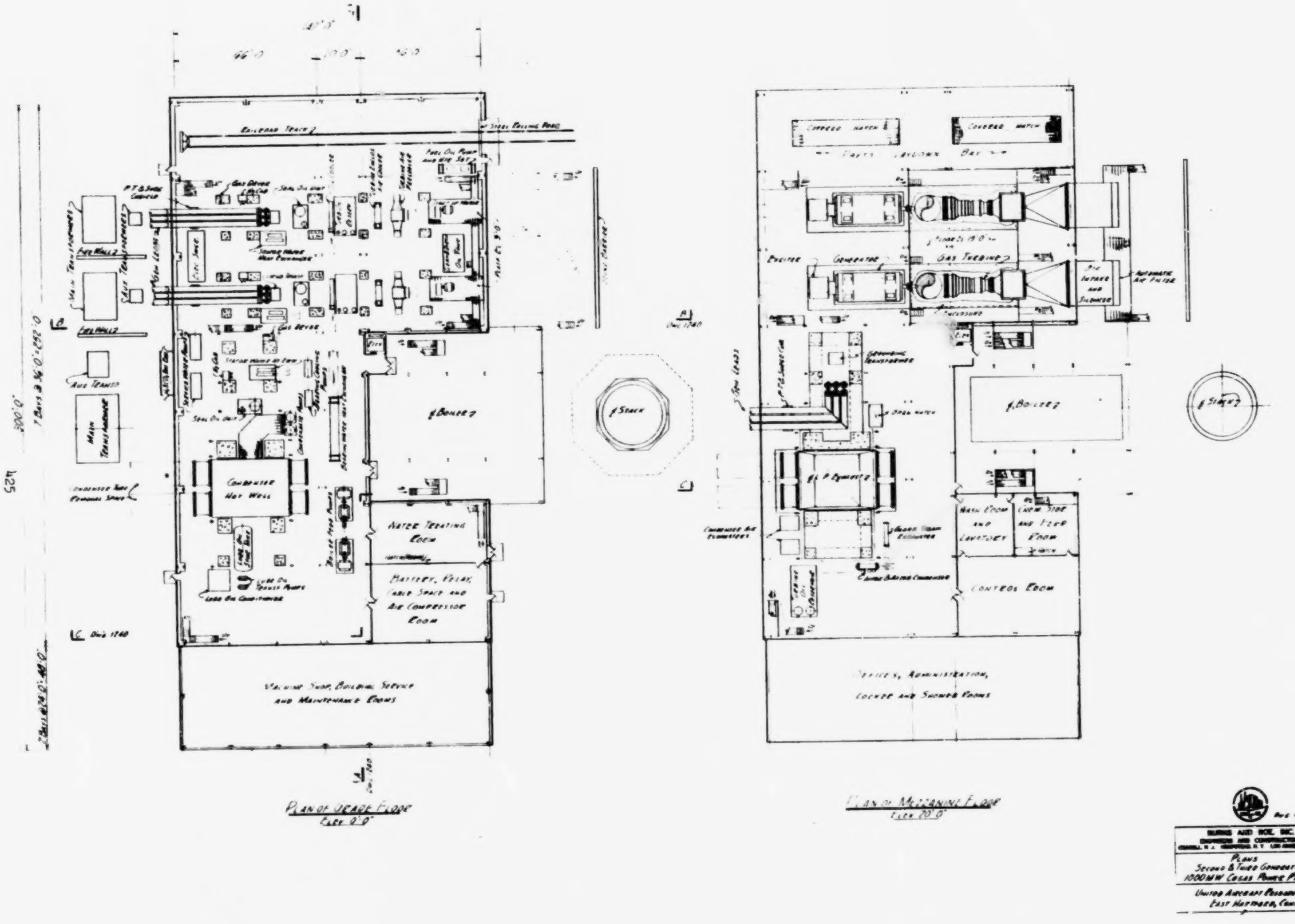


FIG. 130

BUSBAR POWER COSTS FOR COAL BASED POWER STATIONS




Plan of Grade Floor
Elev 0'0"
Plan of Mezzanine Floor
Elev 20'0"
Railroad Track
Steel Rolling Door
Main Transformers
Aux Transformers
Fire Walls
Aux Trans
Main Transformer
Condenser Hot Well
Water Treating Room
Battery, Relay, Cable Space and Air Compressor Room
Machine Shop, Building Service and Maintenance Rooms
Boiler Feed Pumps
Lube Oil Transfer Pumps
Lube Oil Conditioner
Condenser Tube Removal Space
Fuel Oil Pump and Htr Set
Gas Drive
Seal Oil Unit
Elec Space
¢ Boiler
¢ Stack
Covered Hatch
Parts Laydown Bay
Exciter
Generator
Gas Turbine
Air Intake and Silencer
Automatic Air Filter
Grounding Transformer
Open Hatch
¢ L.P. Turbine
Condenser Air Ejectors
Wash Room and Lavatory
Chem Stor and Test Room
Control Room
Offices, Administration, Locker and Shower Rooms
Gen Leads
300'0"
7 Bays @ 36'0" = 252'0"
2 Bays @ 24'0" = 48'0"
46'0"
20'0"
18'0"
84'0"
Dwg 1240
Dwg 1241
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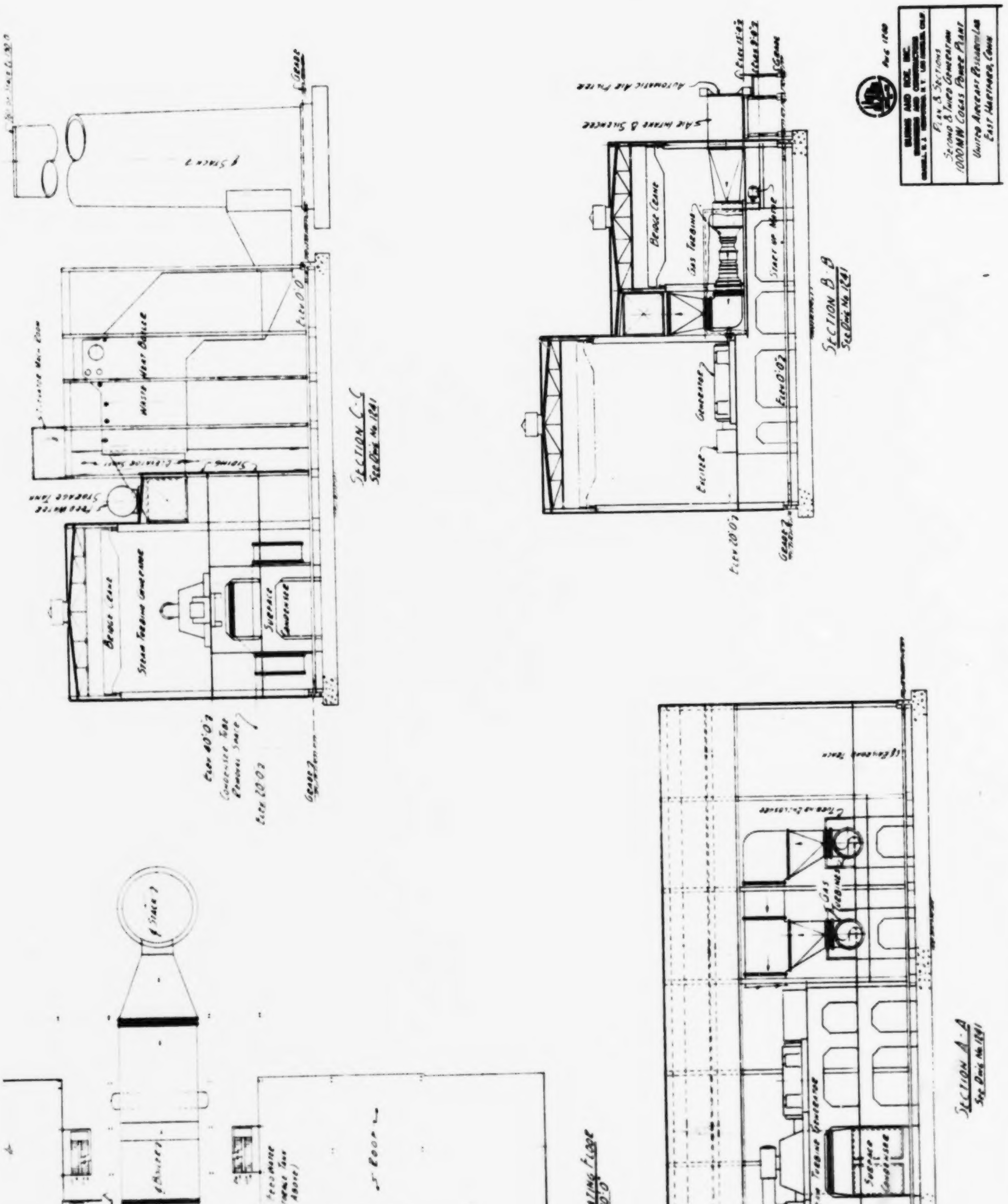
ORADELL, N.J. HEMPSTEAD, N.Y. LOS ANGELES, CALIF.
Plans
Second & Third Generation
1000 MW Cogas Power Plant
United Aircraft Research Lab
East Hartford, Conn

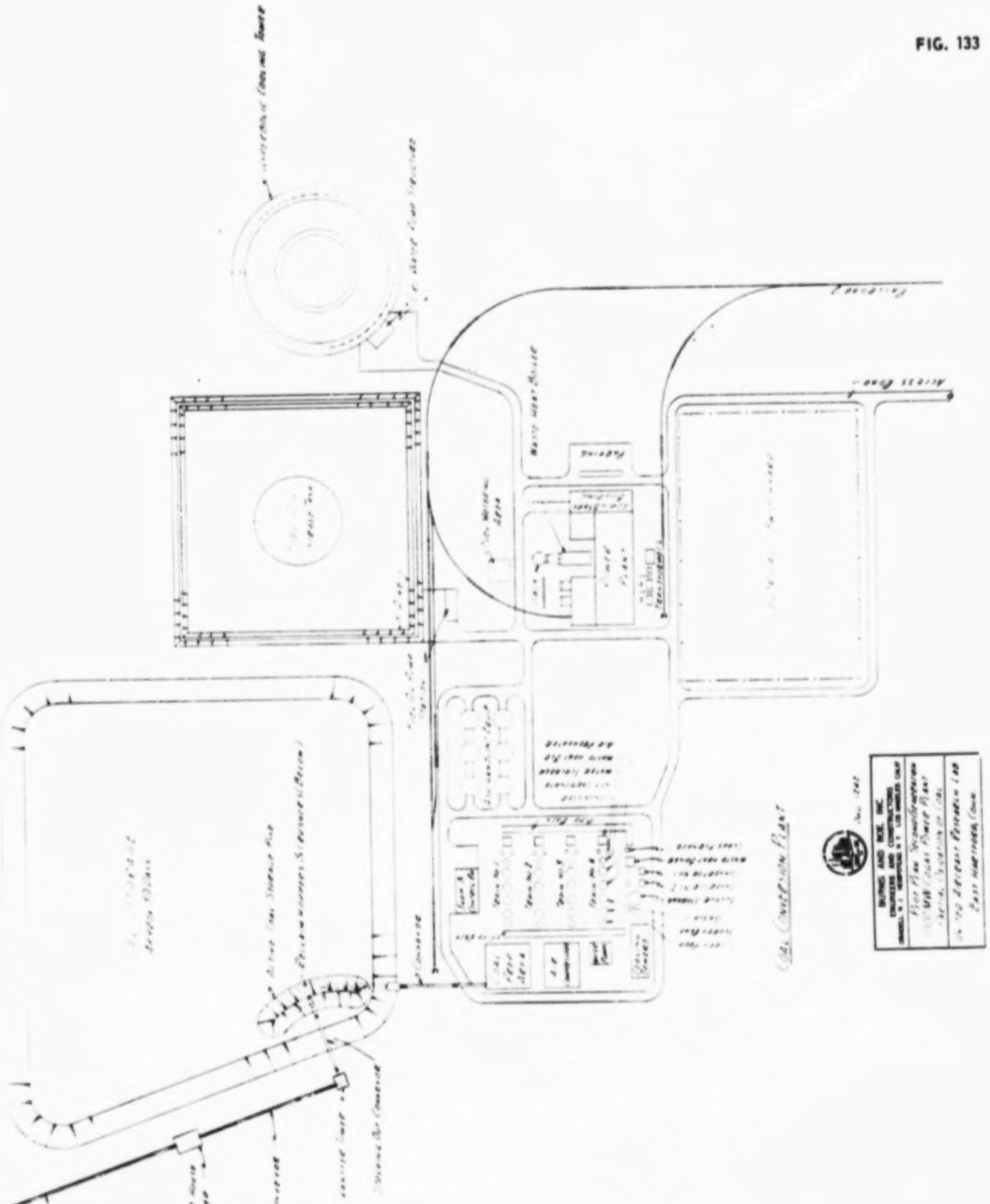
FIG. 131

FIG. 132



Plan of Operating Floor
Elev 40'0"
Section A-A
See Dwg No 1241
Section B-B
See Dwg No 1241
Section C-C
See Dwg No 1241
Steam Turbine Generator
Surface Condenser
Bridge Crane
Waste Heat Boiler
Gas Turbine
Generator
Exciter
Stack
Automatic Air Filter
Air Intake & Silencer
Grade
Elev 0'0"
Elev 20'0"
Elev 40'0"
Burns and Roe, Inc.
Plan & Sections
Second & Third Generation
1000 MW COGAS Power Plant
United Aircraft Research Lab
East Hartford, Conn

FIG. 133


Coal Conversion Plant
BURNS AND ROE, INC.
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Plot Plan
Coal Power Plant
United Aircraft Research Lab
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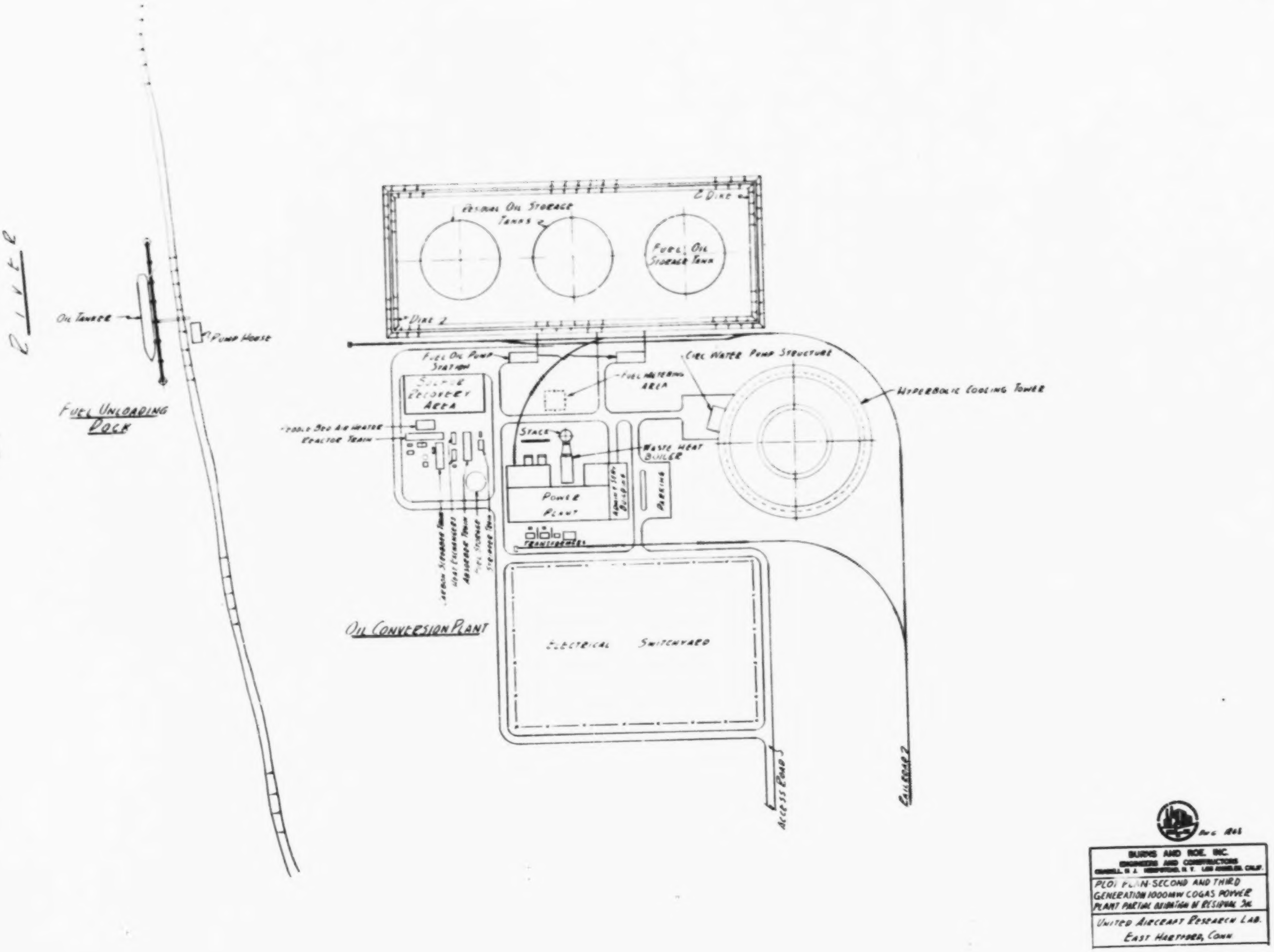
Cooling Tower
Parking
Access Road
Railroad
Power Plant
Conveyor

FIG. 134


River
Oil Tanker
Pump House
Fuel Unloading Dock
Residual Oil Storage Tanks 2
Fuel Oil Storage Tank
Dike 2
Fuel Oil Pump Station
Sulfur Recovery Area
Air Heater
Reactor Train
Carbon Separator Train
Heat Exchangers
Absorber Train
Fuel Storage
Stripper Train
Oil Conversion Plant
Fuel Metering Area
Stack
Waste Heat Boiler
Power Plant
Admin & Serv Building
Parking
Transformers
Circ Water Pump Structure
Hyperbolic Cooling Tower
Electrical Switchyard
Access Road
Railroad
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PLOT PLAN-SECOND AND THIRD GENERATION 1000MW COGAS POWER PLANT PARTIAL OXIDATION OF RESIDUAL OIL
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East Hartford, Conn

BUSBAR POWER COSTS FOR OIL-BASED POWER STATIONS

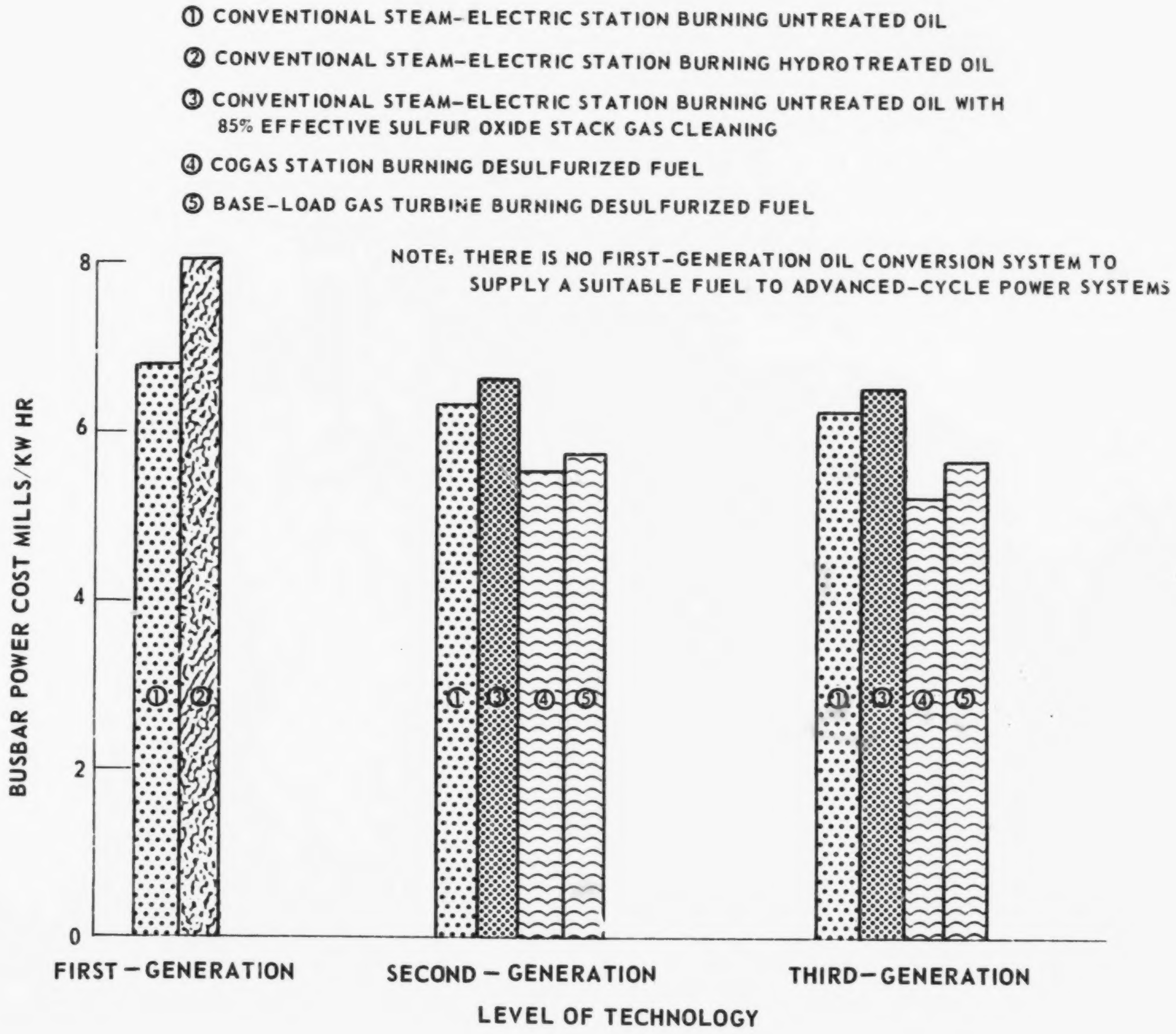


FIG. 135

FIG. 136

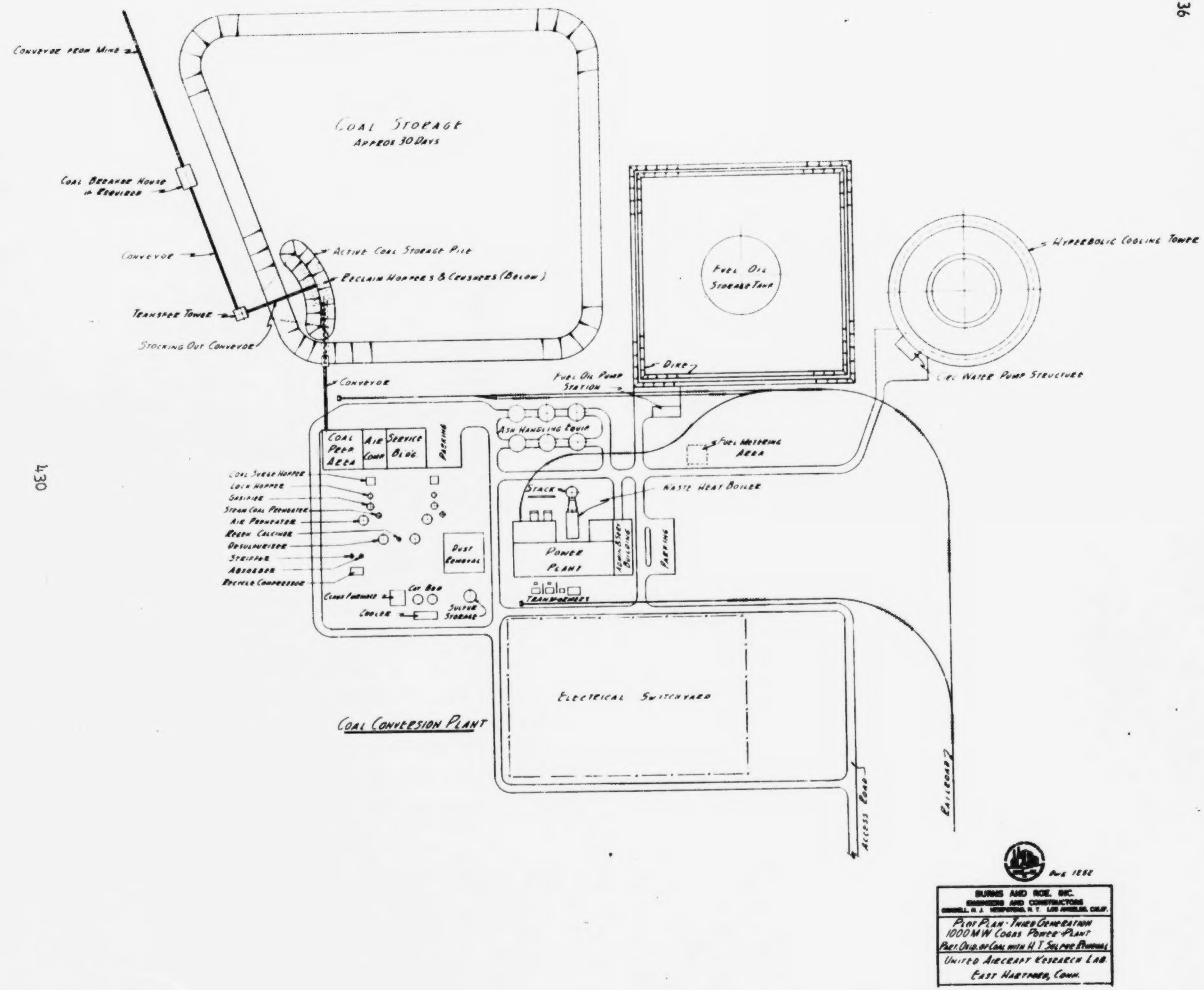
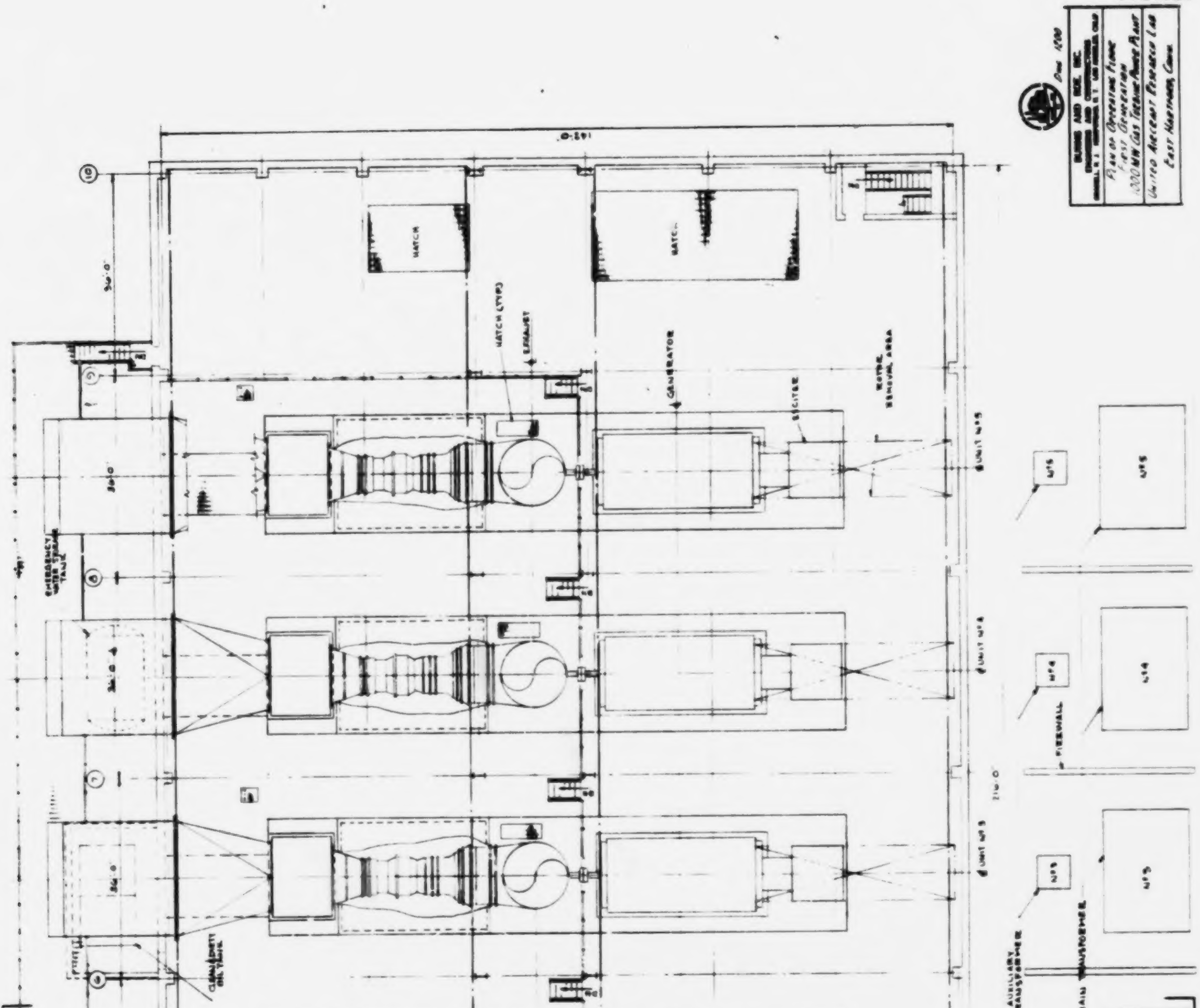
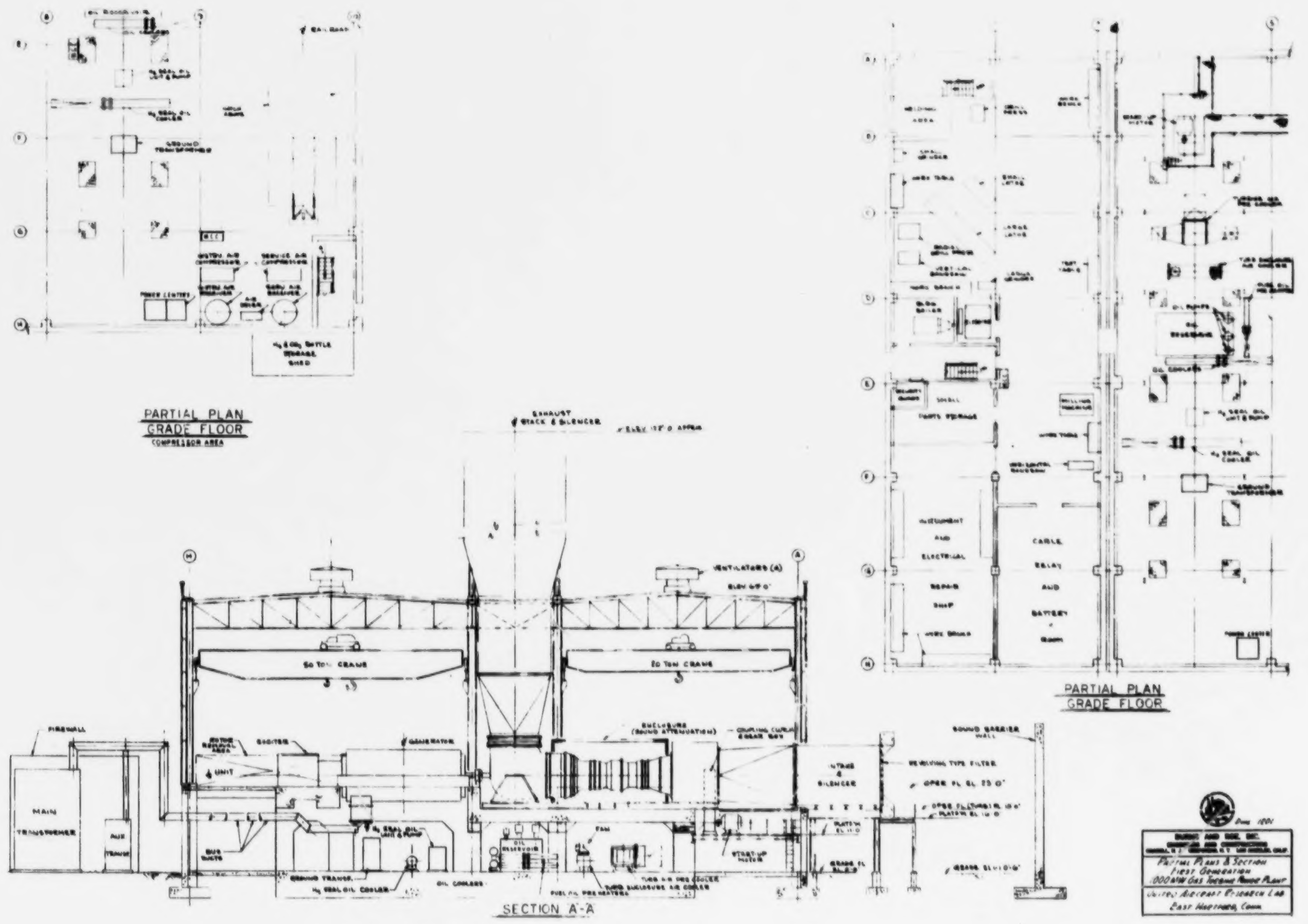

Conveyor from Mine
Coal Breaker House if Required
Conveyor
Transfer Tower
Stocking Out Conveyor
Coal Storage
Approx 30 Days
Active Coal Storage Pile
Reclaim Hoppers & Crushers (Below)
Conveyor
Fuel Oil Storage Tank
Dike
Fuel Oil Pump Station
Hyperbolic Cooling Tower
Circ Water Pump Structure
Ash Handling Equip
Fuel Metering Area
Coal Prep Area
Air Comp
Service Bldg.
Parking
Coal Surge Hopper
Lock Hopper
Gasifier
Steam Coal Preheater
Air Preheater
Regen Calciner
Desulphurizer
Stripper
Absorber
Recycle Compressor
Dust Removal
Claus Furnace
Cat Bed
Cooler
Sulfur Storage
Stack
Waste Heat Boiler
Power Plant
Admin & Serv Building
Parking
Transformers
Electrical Switchyard
Coal Conversion Plant
Access Road
Railroad
Aug 1962
BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL, N.J. HEMPSTEAD, N.Y. LOS ANGELES, CALIF.
Plot Plan - Third Generation
1000 MW Cogas Power-Plant
Part. Oxid. of Coal with H T Sulfur Removal
United Aircraft Research Lab.
East Hartford, Conn.

FIG.137


BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
Plan of Operating Floor
First Generation
1000 MW Gas Turbine Power Plant
United Aircraft Research Lab
East Hartford, Conn.
Dec 1966
HATCH
HATCH (TYP)
EXHAUST
GENERATOR
EXCITER
ROTOR REMOVAL AREA
EMERGENCY WATER STORAGE TANK
CLEAN & DIRTY OIL TANK
REVOLVING TYPE FILTER (TYP)
UNIT NO 1
UNIT NO 2
UNIT NO 3
UNIT NO 4
UNIT NO 5
AUXILIARY TRANSFORMER
MAIN TRANSFORMER
FIREWALL
LOCKER ROOM
SHOWER ROOM
MEN'S TOILET
AIR CONDITIONING & EQUIPMENT ROOM
TOILET
CLOSET
CONTROL ROOM
SEC'Y
PLANT SUPER OFFICE
GENERAL OFFICE
RECORDS & FILES
SHIFT FOREMAN'S OFFICE
FIRST AID ROOM
ENGINEERING OFFICE
OFFICE
CONFERENCE ROOM
A
B
C
D
E
F
G
H
1
2
3
4
5
6
7
8
9
10
24'-0"
48'-0"
36'-0"
143'-0"
210'-0"

431

FIG. 138



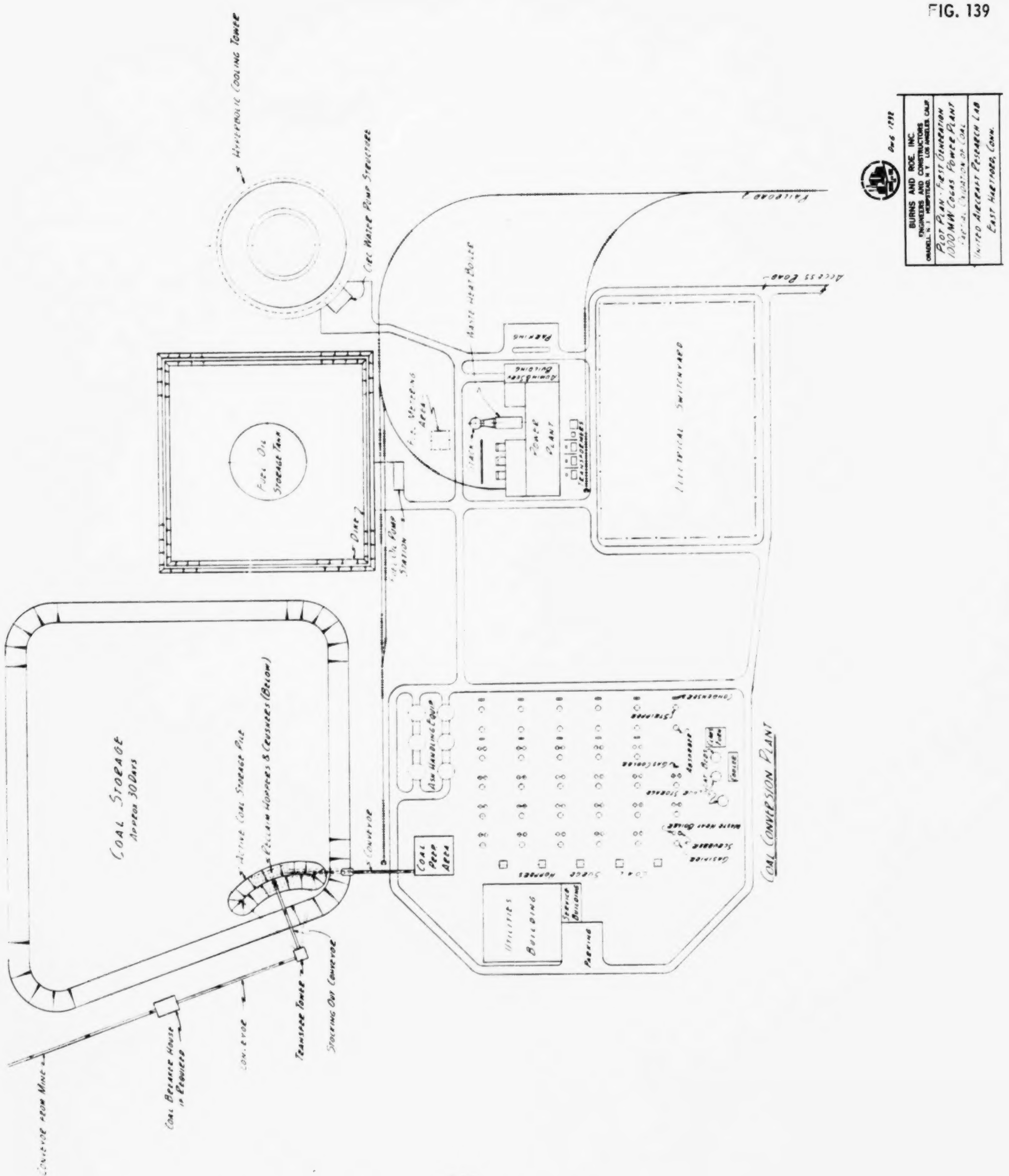
PARTIAL PLAN
GRADE FLOOR
COMPRESSOR AREA

PARTIAL PLAN
GRADE FLOOR

SECTION A-A

Dec 1971

Partial Plans & Section
First Generation
1000 MW Gas Turbine Power Plant
United Aircraft Research Lab
East Hartford, Conn.



Hyperbolic Cooling Tower
Circ Water Pump Structure
Railroad
Waste Heat Boiler
Parking
Running Gear Building
Access Road
Electrical Switchyard
Fuel Oil Storage Tank
Dike
Fuel Oil Pump Station
Fuel Oil Metering Area
Stack
Power Plant
Transformers
Coal Storage
Approx 30 Days
Active Coal Storage Pile
Reclaim Hoppers & Crushers (Below)
Conveyor
Coal Prep Area
Ash Handling Equip
Coal Surge Hoppers
Utilities Building
Service Building
Parking
Gasifier
Scrubber
Waste Heat Boiler
Storage
Gas Cooler
Absorber
Stripper
Condenser
Coal Conversion Plant
Conveyor from Mine
Coal Breaker House if Required
Conveyor
Transfer Tower
Stocking Out Conveyor
Aug 1973
BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL, N.J. HEMPSTEAD, N.Y. LOS ANGELES, CALIF
Plot Plan - First Generation
1000 MW Cogas Power Plant
Partial Oxidation of Coal
United Aircraft Research Lab
East Hartford, Conn.

FIG. 139

FIG. 140

EFFECT OF LOAD FACTOR ON BUSBAR POWER COST

FIRST-GENERATION POWER STATIONS

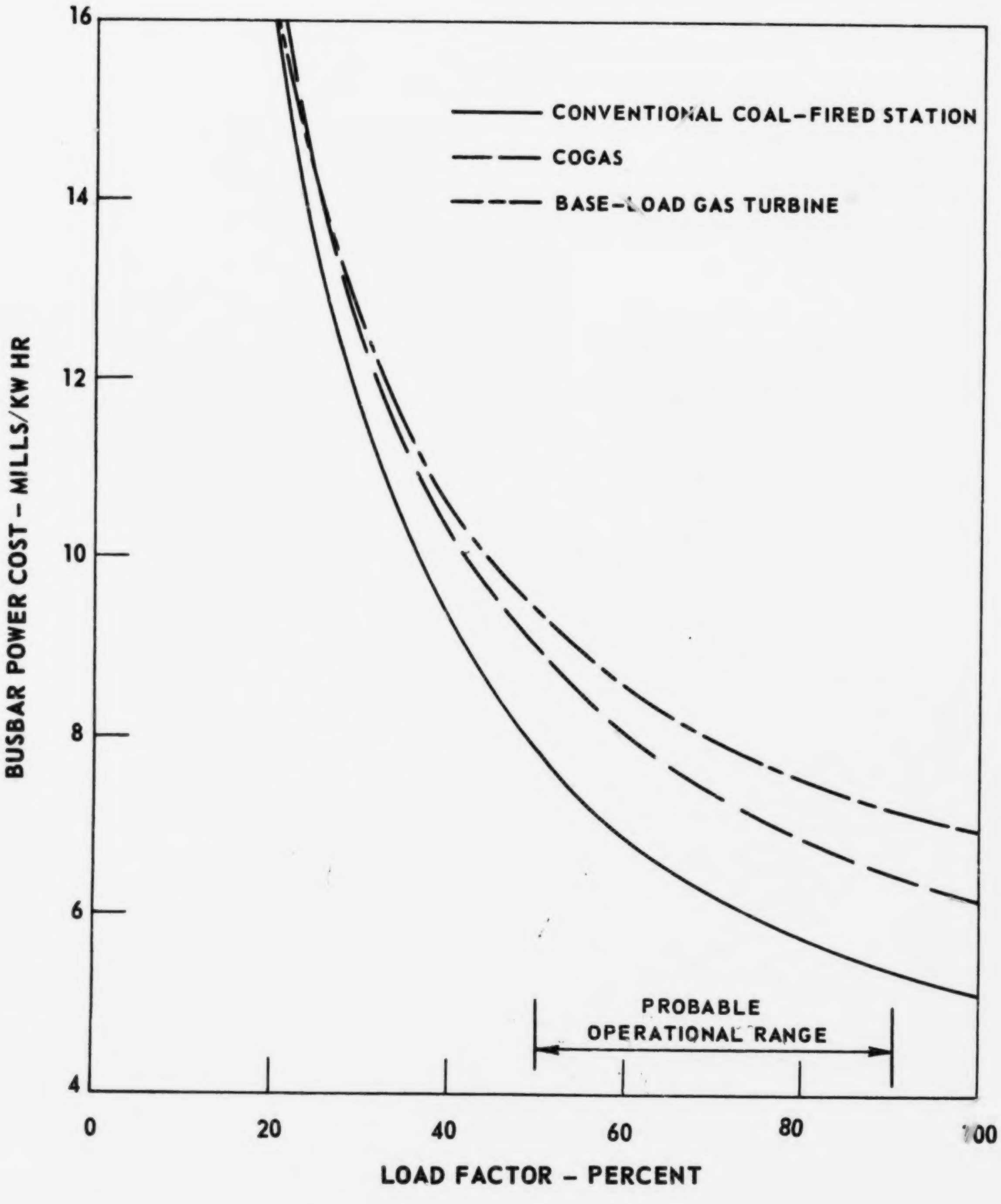
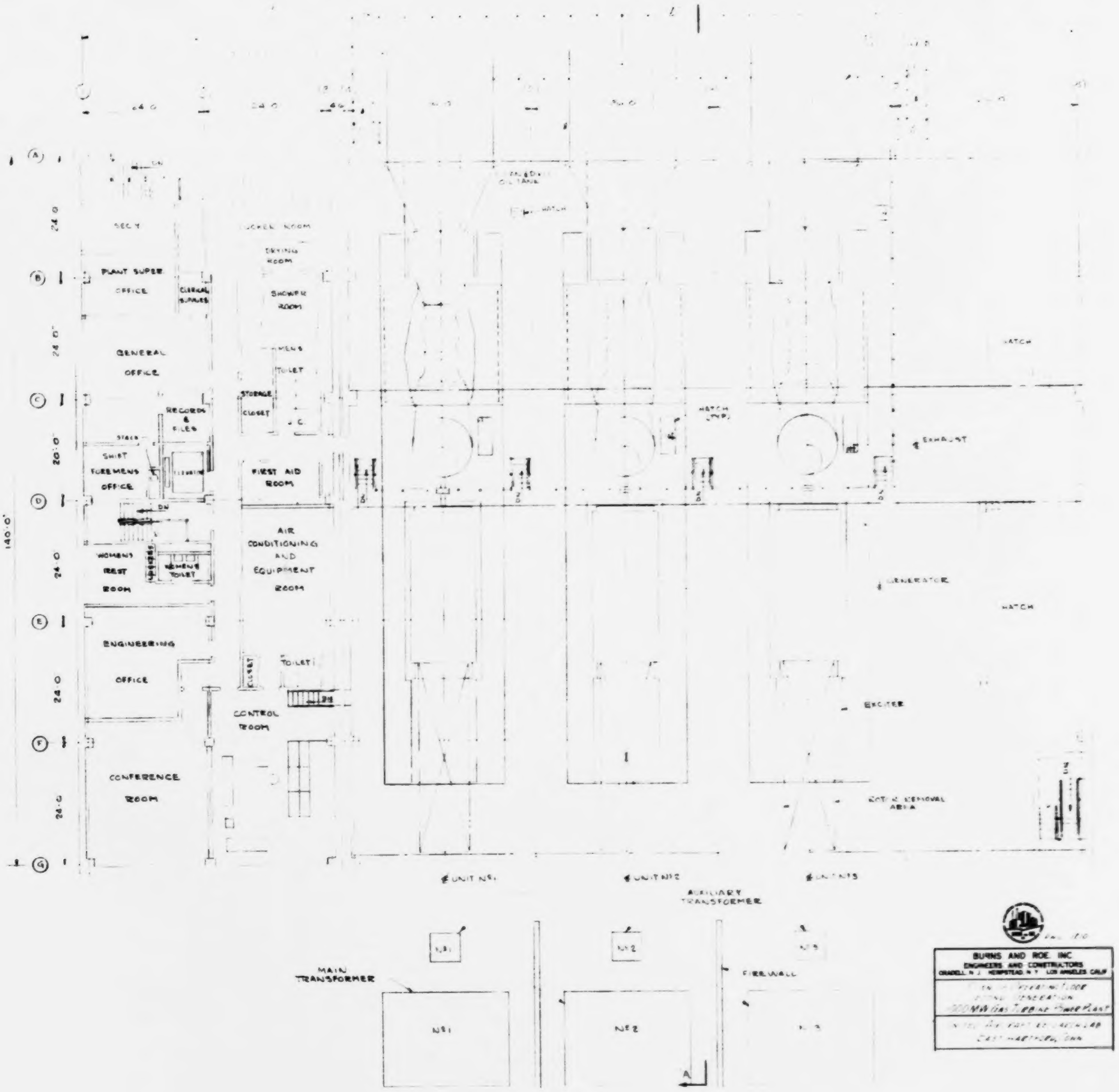


FIG. 141



NOT REPRODUCIBLE

FIG. 142

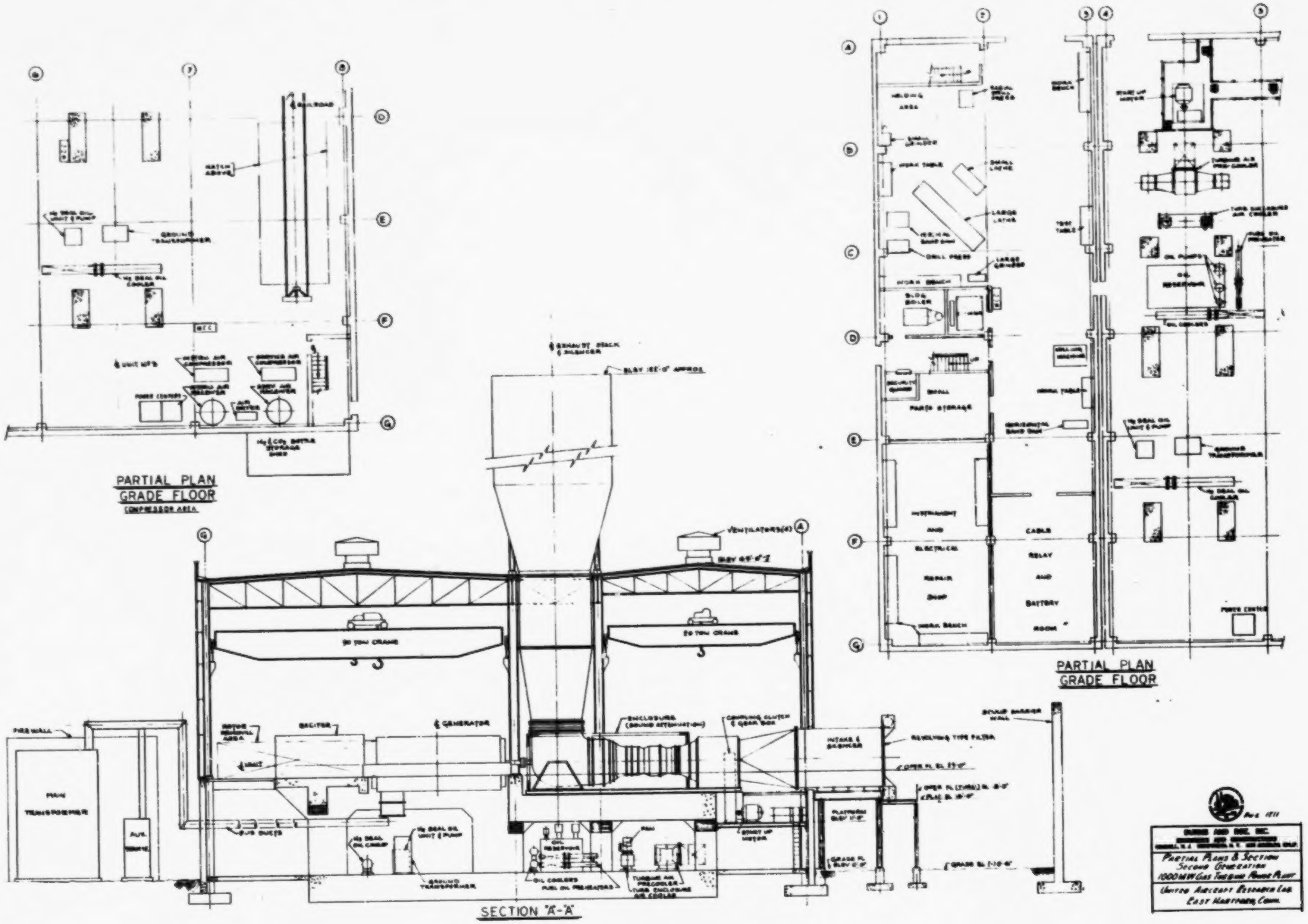
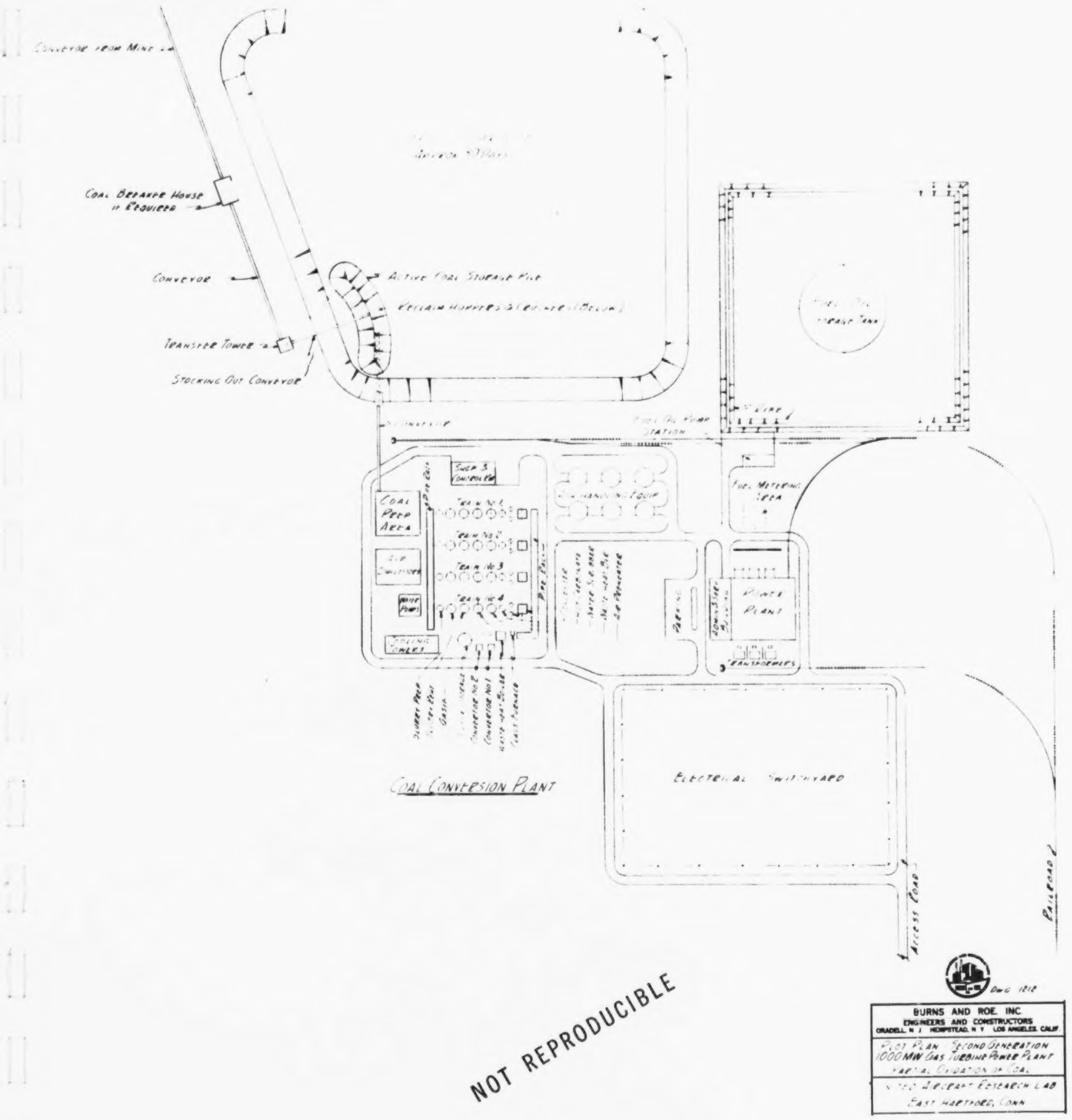
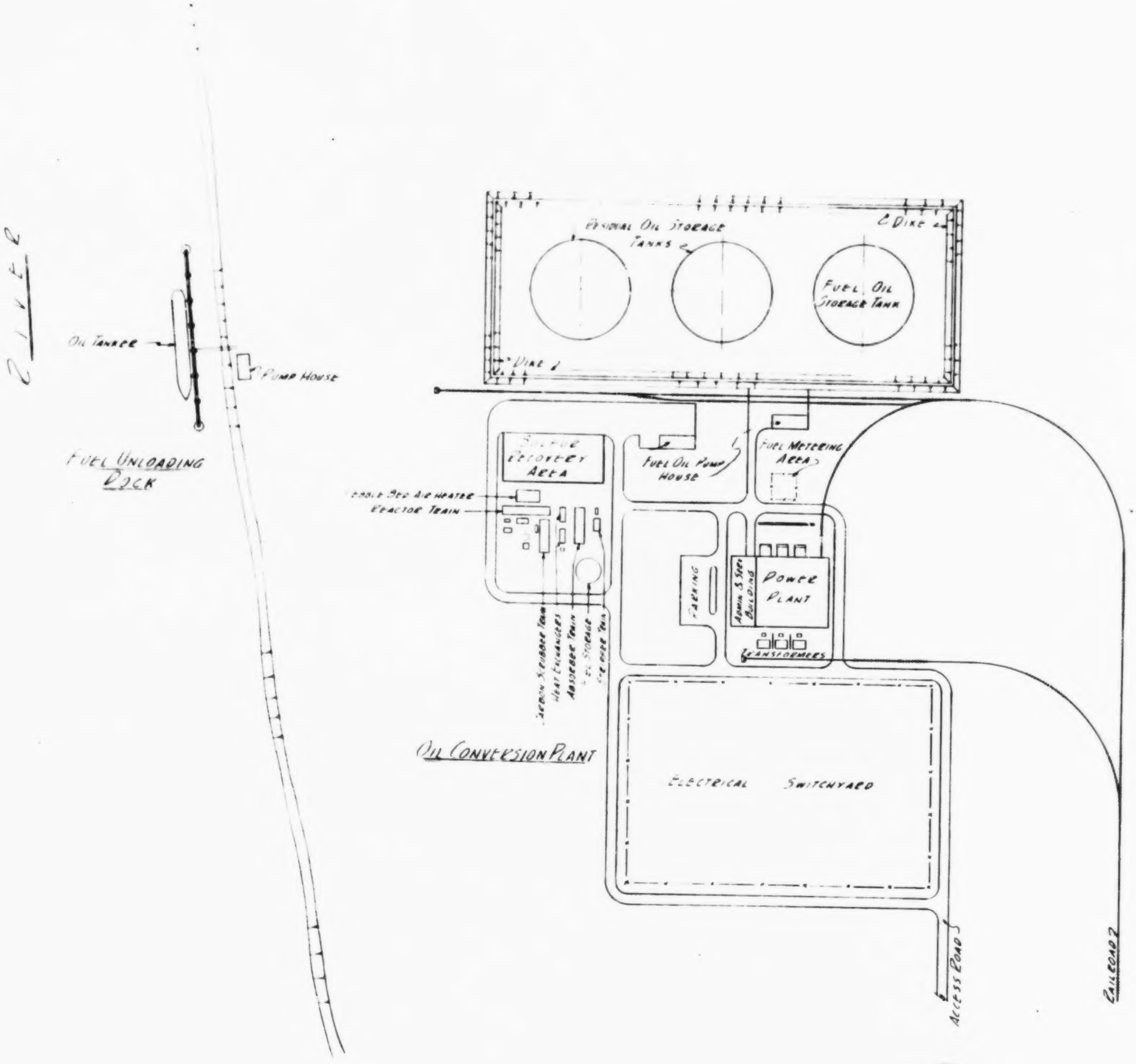

PARTIAL PLAN
GRADE FLOOR
PARTIAL PLAN
GRADE FLOOR
COMPRESSOR AREA
SECTION "A"-"A"
HOLDING AREA
RADIAL DRILL PRESS
SMALL GRINDER
WORK TABLE
SMALL LATHE
LARGE LATHE
VERTICAL BAND SAW
DRILL PRESS
LARGE GRINDER
WORK BENCH
BLDG. BOILER
SECURITY GUARD
SMALL PARTS STORAGE
HORIZONTAL BAND SAW
WORK TABLE
TEST TABLE
WORK BENCH
START UP MOTOR
TURBINE AIR PRECOOLER
TURB. ENCLOSURE AIR COOLER
FUEL OIL PREHEATER
OIL PUMPS
OIL RESERVOIR
OIL COOLERS
H2 SEAL OIL UNIT & PUMP
GROUND TRANSFORMER
H2 SEAL OIL COOLER
INSTRUMENT AND ELECTRICAL REPAIR SHOP
WORK BENCH
CABLE RELAY AND BATTERY ROOM
POWER CENTER
RAILROAD
HATCH ABOVE
MCC
UNIT NO 2
INSTRUMENT AIR COMPRESSOR
SERVICE AIR COMPRESSOR
POWER CENTERS
INSTRU. AIR RECEIVER
AIR DRYER
SERV. AIR RECEIVER
H2 & CO2 BOTTLE STORAGE SHED
EXHAUST STACK & SILENCER
ELEV 122'-0" APPROX.
VENTILATORS(4)
ELEV 45'-0"
20 TON CRANE
50 TON CRANE
FIRE WALL
MAIN TRANSFORMER
AUX. TRANSF.
ROTOR REMOVAL AREA
UNIT
EXCITER
GENERATOR
BUS DUCTS
H2 SEAL OIL COOLER
H2 SEAL OIL UNIT & PUMP
GROUND TRANSFORMER
OIL RESERVOIR
OIL COOLERS
FUEL OIL PREHEATERS
FAN
TURBINE AIR PRECOOLER
TURB. ENCLOSURE AIR COOLER
START UP MOTOR
ENCLOSURE (SOUND ATTENUATION)
COUPLING CLUTCH & GEAR BOX
INTAKE & SILENCER
REVOLVING TYPE FILTER
OPER. FL. EL. 23'-0"
PLATFORM ELEV 17'-0"
GRADE FL. ELEV 0'-0"
GRADE EL. (-)10'-6"
SOUND BARRIER WALL
A
G
1
2
3
4
5
6
7
8
A
B
C
D
E
F
G
BURNS AND ROE, INC.
PARTIAL PLANS & SECTION
SECOND GENERATION
1000 MW GAS TURBINE POWER PLANT
UNITED AIRCRAFT RESEARCH LAB.
EAST HARTFORD, CONN.
Aug 1971

FIG. 143



NOT REPRODUCIBLE

FIG. 144



Dwg 1813

BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL, N. J. HEMPSTEAD, N. Y. LOS ANGELES, CALIF.

Plot Plan - Second Generation
1000 MW Gas Turbine Power Plant
Partial Oxidation of Residual Oil

United Aircraft Research Lab.
East Hartford, Conn.

EFFECT OF LOAD FACTOR ON BUSBAR POWER COST

SECOND-GENERATION POWER STATION

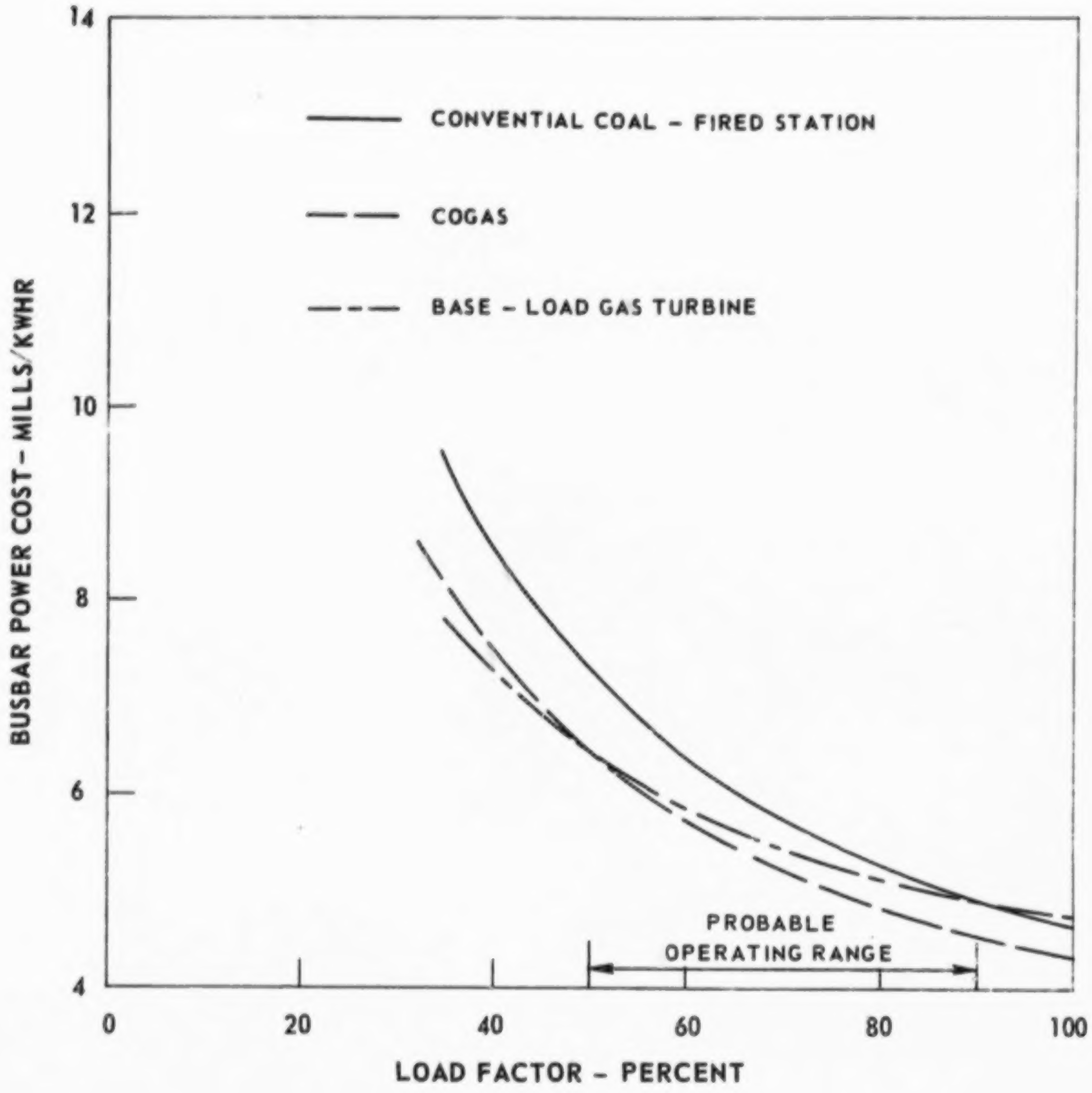
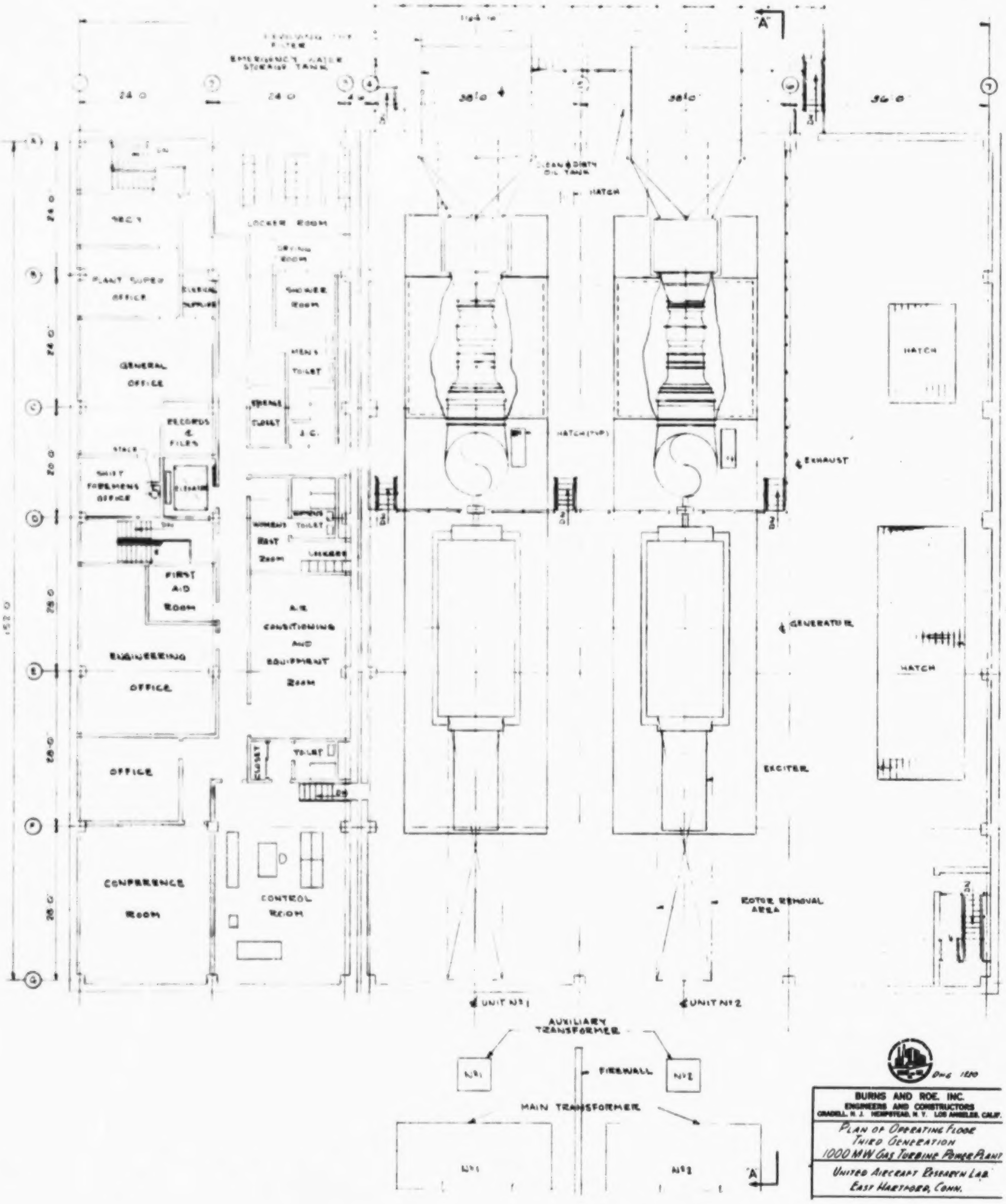


FIG. 146



BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL, N. J. HEMPSTEAD, N. Y. LOS ANGELES, CALIF.

PLAN OF OPERATING FLOOR
THIRD GENERATION
1000 MW GAS TURBINE POWER PLANT

UNITED AIRCRAFT RESEARCH LAB.
EAST HARTFORD, CONN.

FIG. 147

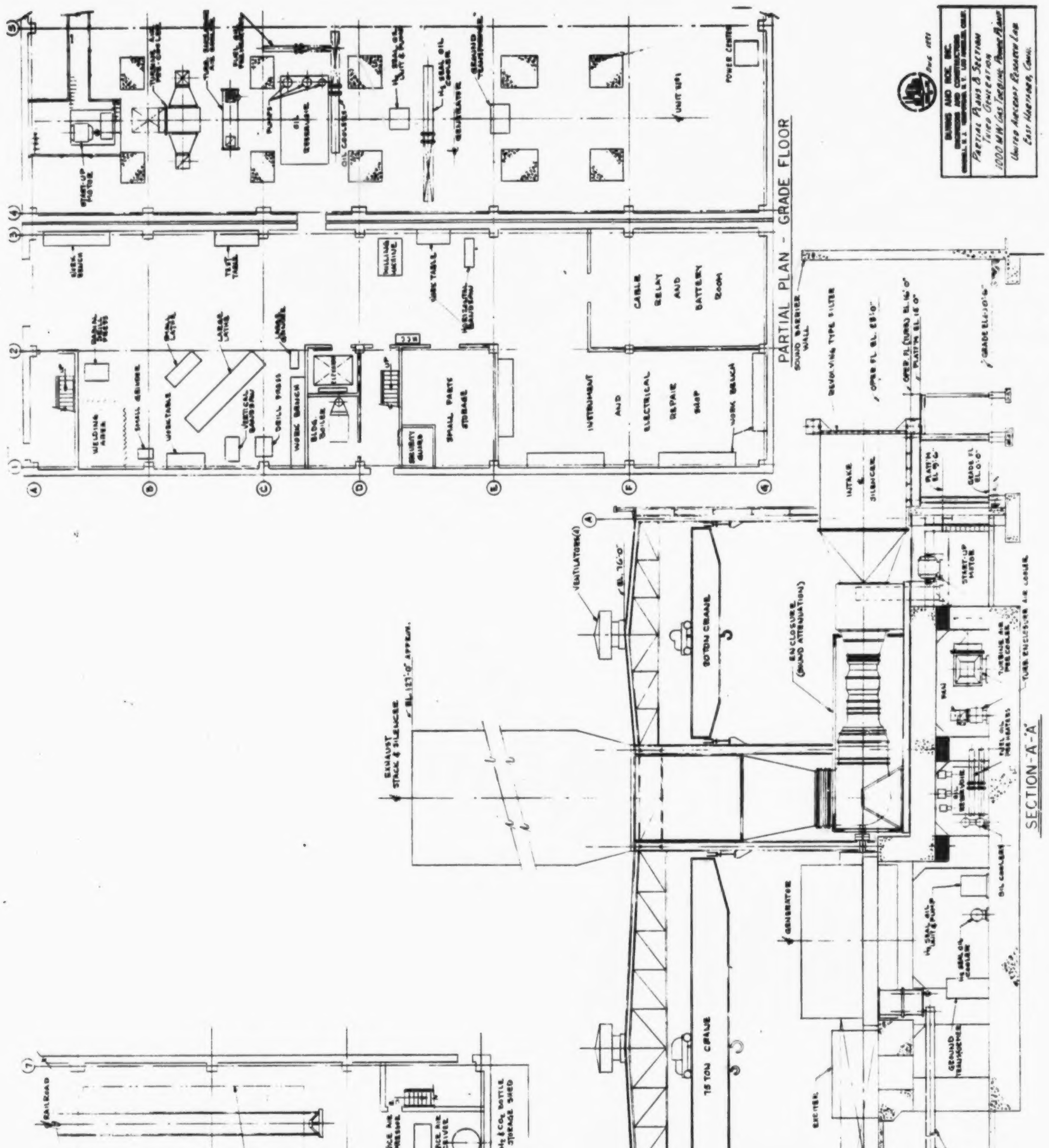
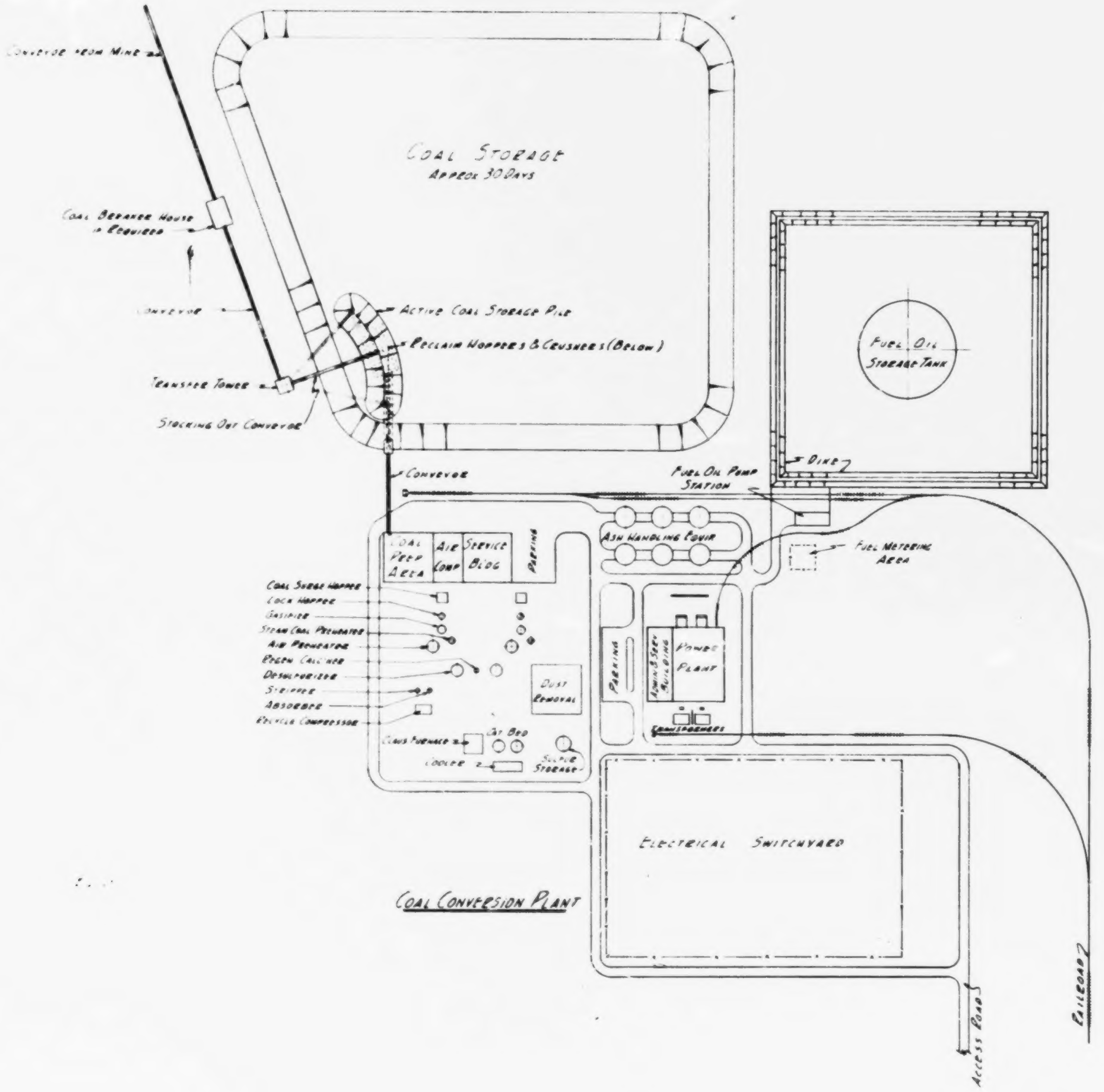

BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
PARTIAL PLANS & SECTION
TURBO GENERATION
1000 MW GAS TURBINE POWER PLANT
UNITED AIRCRAFT RESEARCH LAB
EAST HARTFORD, CONN.
PARTIAL PLAN - GRADE FLOOR
SECTION 'A-A'
PARTIAL PLAN
GRADE FLOOR
COMPRESSOR AREA
WELDING AREA
SMALL GRINDER
DRILL PRESS
WORK BENCH
SMALL PARTS STORAGE
INSTRUMENT AND ELECTRICAL REPAIR SHOP
CABLE RELAY AND BATTERY ROOM
START-UP MOTOR
OIL COOLERS
GENERATOR
GROUND TRANSFORMER
UNIT Nº1
UNIT Nº2
POWER CENTER
SOUND BARRIER WALL
REVOLVING TYPE FILTER
INTAKE & SILENCER
ENCLOSURE (SOUND ATTENUATION)
EXHAUST STACK & SILENCER
VENTILATORS
80 TON CRANE
75 TON CRANE
EXCITER
BUS DUCTS
AUX. TRANSF.
MAIN TRANSFORMER
FIREWALL
RAILROAD
HATCH ABOVE
SERVICE AIR COMPRESSOR
INSTRU. AIR COMPRESSOR
H_2 & CO_2 BOTTLE STORAGE SHED

FIG. 148



Dwg 1228

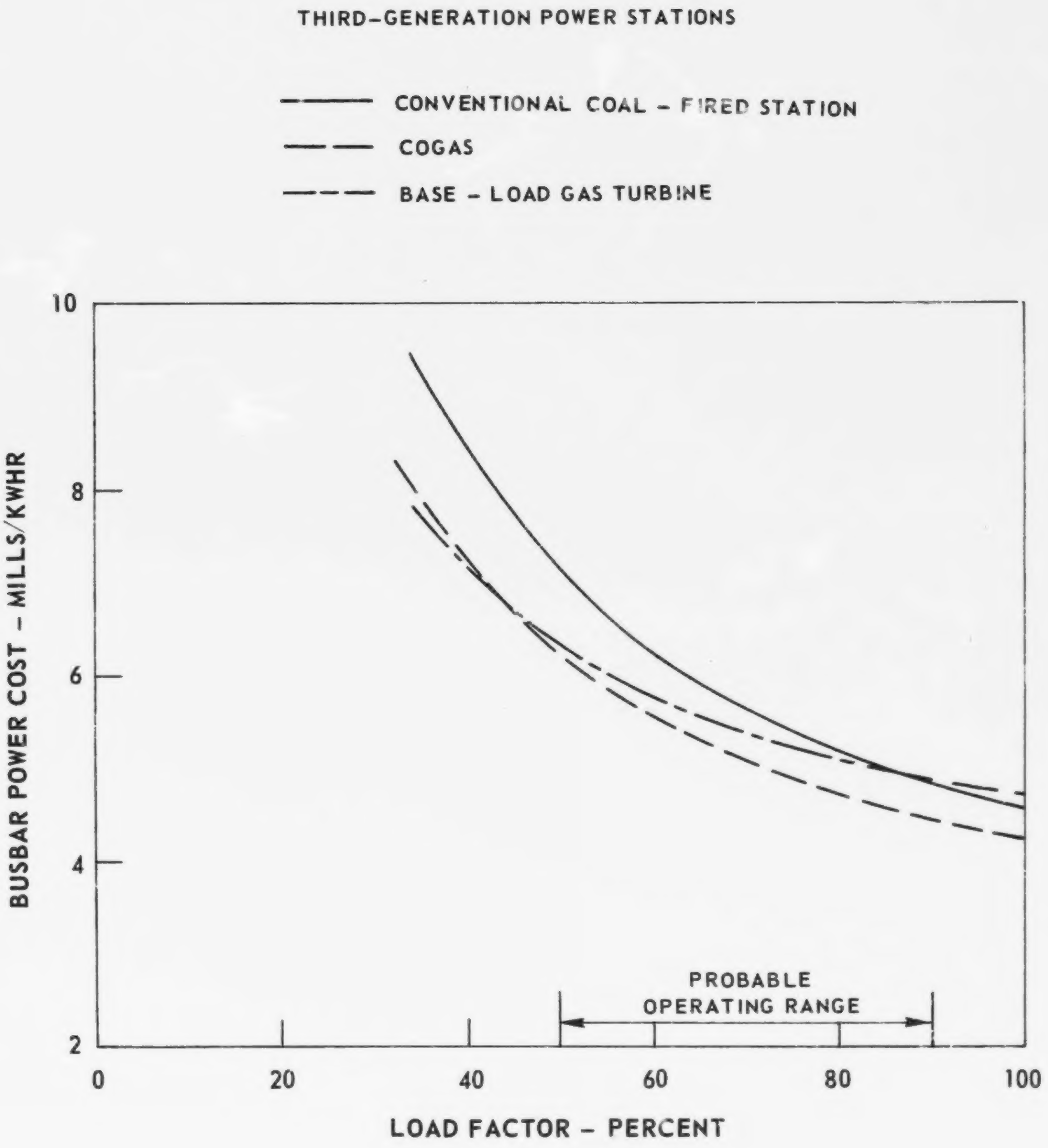
BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
ORADELL, N. J. HEMPSTEAD, N. Y. LOS ANGELES, CALIF.

Plot Plan - Third Generation
1300 MW Gas Turbine Power Plant
Part. Oxid. of Coal with H. T. Sulfur Removal

United Aircraft Research Lab
East Hartford, Conn.

FIG. 149

EFFECT OF LOAD FACTOR ON BUSBAR POWER COSTS



APPENDICES

APPENDIX A

DESCRIPTION OF REPRESENTATIVE POWERPLANT LOCATIONS

In accordance with the Contract Work Statement (Ref. 178), power system configurations were considered for two locations: (1) a coal-fired 1000-Mw station in the Great Lakes area, and (2) an oil-fired 1000-Mw station in the northeastern United States.

Great Lakes Area

The coal-fired plant was assumed to be located at the site of the mine supplying fuel. This limited the Great Lakes location to essentially two areas, southern Indiana and southern Illinois. Because of the readily accessible data on Illinois coal (see Appendix C), a southern Illinois location was selected.

The actual location of the site was assumed to resemble that of the Lake Coffeen station of the Central Illinois Public Service Company (Ref. 179). This station is situated about 8 miles southwest of Hillsboro, Illinois, essentially on top of the Herron Seam (or Seam No. 6) of Illinois. While the actual Coffeen station is supplied cooling water from a man-made lake, it was assumed that the hypothetical station considered herein would be supplied cooling tower makeup water from local, natural supplies.

The plant would be located in a very rural area, and it was assumed that sufficient land would be available for locating the fuel processing plant, power station, and a 90-day above-ground coal supply at a reasonable cost. No unusual powerplant foundations or other unique site preparations were assumed. The power plant would be fully enclosed as befits a station in this climate.

Northeastern United States

A wider choice of location was available for the oil-fired plant in the northeastern United States. The only requirements to be met were for a supply of adequate cooling tower makeup water and access to a waterway which would allow oil delivery at reasonable cost. The actual site selected is on the Hudson River, near Roseton, New York. Burns and Roe., Inc., is currently involved in the construction of a power plant at this location.

This site is in a semirural area and required that some consideration be given to aesthetics. Thus the station, while fully enclosed as would be the coal-fired station, would have higher construction costs. In addition, the actual Roseton station required extensive foundation work because of the nature of the site. To reflect this, two station costs have been developed, one accounting for the extra foundation work, and one neglecting this factor.

The river location would allow oil to be brought directly to the station by small tanker. A storage facility sufficient for 30 days supply of oil was assumed.

APPENDIX B

TRANSPORTATION COSTS FOR FUEL

A basic assumption of the study contained in the body of this report was that the power plant would be located at the source of the raw fuel; i.e., a mine mouth location for the station using coal, and extensive on-site storage facilities for the station using oil. This assumption allowed the busbar cost of power to be determined on an equable basis for the various stations without introducing the confusion of fuel transportation charges. It is of interest, however, to establish some estimates of fuel transportation charges since it is very likely that all future power stations will not be located at the raw fuel source.

Coal Transport

The most promising coal conversion and desulfurization processes evaluated produce a low-heat-content gaseous fuel. There exist a number of alternative configurations for the location of the fuel-cleanup facility, the power generating facility, the mine, and the ultimate consumers. Three of the more obvious configurations are:

1. Both the fuel cleanup facility and the power plant could be located at the mine mouth, a configuration which would permit integration of the fuel cleanup process with the power cycles and eliminate any major coal transportation costs.

2. The gasification facility could be located at the mine mouth and the product fuel then transmitted to a power-generating facility situated at a distance. As an alternative which would increase the fuel gas transmission efficiency, the low-Btu gas could be methanated to produce a somewhat higher heating-value fuel.

3. An integrated gasification facility and power plant could be situated at a distance from the mine. Coal would then be transported from the mine to this integrated facility using conventional transportation systems.

Clearly, the first configuration would produce the lowest fuel gas costs, since there would be no fuel transportation charges. However, the overall cost of power transmission from the mine-mouth power plant to the ultimate consumers might outweigh the fuel transportation savings. Consequently, determination of the most desirable location for the power-generating facility with relation to coal deposits and ultimate consumers would require further study.

A stipulation to locate the power-generating facility away from the mine mouth would necessitate one of the remaining two configurations. In the second configuration, the coal would be gasified at the mine mouth with subsequent pipeline transmission to the power-generating facility. However, this system is intuitively unattractive for two outstanding reasons. First, the use of a remote gasification facility would prevent any integration between the fuel cleanup process and the power cycle, e.g., the high temperature and pressure of the product fuel resulting from the gasification process both could be forfeited as a result of pipeline transmission. Second, the reported low costs for the pipeline transmission gas at high load factors would not apply to the transmission of coal gasified by the processes considered since, if the gas were not methanated, the heating value of these fuels would be less than 200 Btu/ft^3, compared to the approximately 1000 Btu/ft^3 heating value of conventional pipeline gas. Therefore, the transmission of this low-heating-value fuel easily could be five times more expensive on a Btu basis than the transmission of conventional pipeline gas. (The transmission cost of natural gas is given in Ref. 180 as approximately 2¢/million Btu-100 mi and the distribution cost is given as approximately 4¢/million Btu-100 mi.)

The third configuration appears intuitively more desirable than the second. Integration of the gasification facility with the power plant allows overall credits for the elevated temperature and pressure of the gasification products. The fuel could be transported by rail as raw coal between the mine and the gasification-power plant complex, and the resulting cost could be considerably less than that for transmission of a low-Btu gas, particularly if existing rail systems can be used. In addition, the cost of coal transport by rail recently has been reduced significantly by the introduction of new high-speed loading and unloading techniques and by the use of single-purpose cars. These highly efficient "unit trains" are approximately five time more productive than conventional trains (Ref. 181), and their overall economies have brought about a renaissance in energy transport by rail.

A quantitative comparison of the second and third configurations can be made by comparing transportation costs of fuel gas and coal. The results of a survey of the use of unit trains for coal transport are shown in Table 107, and representative coal transport costs by rail are shown in Fig. 150. Since the Btu content of the gasified coal, with its high concentration of inert gases, would be approximately one-fifth that of conventional pipeline gas, the transmission costs for the gasified coal would be approximately 10¢/million Btu-100 mi (based on a transmission cost of 2%/million Btu-100 mi for conventional gas transmission). A comparison of this cost with that for the rail transport of coal shown in Fig. 150 confirms the conclusion that, in this situation, energy movement by rail is comparatively cheaper. Consequently, an integrated coal gasification-power plant facility with rail transport of coal appears to be the most desirable configuration for the production of electricity from coal at a location away from the mine mouth.

As noted in the description of the second configuration, an alternative would be to produce a 1000-Btu/ft^3 fuel gas from coal and transmit this to power plants located at distances from the mine. The results of an evaluation of this alternative are shown in Fig. 151. The busbar power cost for a 1000-Mw COGAS station using 1980 technology is shown to be higher for a station using the 1000-Btu/ft^3 gas when compared to an identical station using a low-heat-content fuel gas produced on-site from coal shipped by rail and then gasified. This higher cost of high-Btu gas is due to several factors: an additional capital cost of the station (about $7.7/kw) due to the increased cost of turbo-machinery to compress the low-pressure 1000-Btu/ft^3 fuel piped to the power plant; the difference in the base cost of fuel, assumed as 40¢/million Btu for pipeline gas versus 38¢/million Btu for the low-Btu fuel; and, most importantly, the difference in transportation costs.

Oil Transportation

Vacuum gas oil hydrodesulfurization would reduce the sulfur content of residual oil to a level which meets first-generation requirements, and theoretically the treatment facility could be located either in Venezuela or on the east coast. In practice, oil importers have found it more economical to desulfurize prior to delivery to the east coast. The fuel oil used by utilities in the New York metropolitan area is delivered by tanker to an off-loading site, whence it is distributed to individual utilities by barge. There is nothing to suggest that the distribution of high-sulfur residual oil would introduce any transportation economies, and there is no instance of a utility engaged in the desulfurization of its own residual fuel. Therefore, for the first generation, use of the existing delivery system would appear to be indicated.

The primary product of the second- and third-generation conversion and desulfurization processes would be a gaseous fuel, and consequently the fuel processing facility must be situated near the ultimate user. When intended for eventual gasification, high-sulfur residual oil would be shipped to the east coast, where it could be distributed to utilities using either of the following two systems:

1. The high-sulfur residual oil could be gasified in a large single facility at the point of tanker unloading with the resulting gaseous fuel delivered to the powerplants via pipeline.

2. The high-sulfur residual oil could be delivered to the individual power plants by barge, rail, truck, or pipeline and gasified at the point of use in a combined gasification-power generation system.

Several points militate against the first option. First, there are no outstanding economies of scale associated with gasification at a facility serving power plants greater than 1000 Mw, since multiple gasification units would be required in any case. Second, the product credits for heat and compression possible in the gasification process would be lost in trasmission if the gasification facility were not situated adjacent to the power plant. Third, the installation of several new large-diameter delivery pipelines from a central depot to scattered power plants in a large metropolitan area would be highly impractical. Land costs and the safety difficulties severely discourage this possibility.

Therefore, it appears that the second option, namely distributing the residual oil prior to treatment with conversion and desulfurization at the point of use, is the only practical alternative. The resulting fuel delivery pattern would resemble the present system, with the difference that the future power facilities would have a gasification capability integrated into the power cycle. The options for the oil delivery appear to be truck, rail, or barge, with oil pipelines in the metropolitan area being eliminated for much the same reason as gas pipelines, noted above. The costs for fuel delivery by truck, rail, and barge are presented in Fig. 152. From these fuel oil delivery costs, it is clear that barging, where possible, is significantly less expensive than truck or rail transport. Therefore, it appears that the most promising system configuration for second- and third-generation periods would utilize barge delivery of untreated residual oil directly to power plants, with the conversion and desulfurization processes operating in conjunction with the power cycle.

APPENDIX C

SELECTION OF FUELS

The following discussion gives the rationale behind the selection of the basic fuels considered and contains tabulations of the analyses of the selected fuels.

Coal

The Great Lakes areas of the United States (Illinois, Indiana, Michigan, and Ohio) was selected as a potential site for a coal processing plant. In 1967, coal consumption for electrical generation in this area was 91 million tons (Ref. 184) compared to the national usage of 293 million tons (Ref. 185). If this same ratio of Great Lakes Region-to-national usage were assumed for the future, 170 million tons of coal would be consumed annually for electrical generation in the Great Lakes Area by 1980 (Ref. 186).

The most widely used steam coals in the Great Lakes area are mined in the midwest (Illinois, Indiana, and western Kentucky), and in Appalachia (Pittsburgh, Kittanning, and Freeport seams of Pennsylvania, Ohio, eastern Kentucky, and West Virginia). Recoverable reserves of these coals are estimated in Ref. 187 to be more than 400 billion tons. The midwest coals are highly volatile B and C bituminous, in which sulfur is predominately organic; that is, sulfur is intimately contained in the coal molecule rather than being present in inorganic forms such as pyrite. Just the reverse in true in many coals found in the Appalachian area. Coals found in both areas are generally agglomerating and have the following characteristics: heating values ranging from 12,000 to 14,500 Btu/lb on a moisture- and ash-free (MAF) basis; fixed carbons ranging from 45% to 55% on an as-receive (AR) basis; volatile matter ranging from 30% to 40% AR; moisture content ranging from 5% to 15% AR; and ash contents ranging from 5% to 20% AR (Ref. 187). Any process for deriving clean fuel from coal should have the capability to process all of these coals. Based on reported gasification yeilds for various coals (Ref. 188), it is believed that selection of one specific coal among the typical highly volatile A, B, or C bituminous steam coals mined in the eastern part of the United States would not significantly affect the generality of the computed economics of a coal conversion process. For this reason, a typical highly volatile B bituminous coal, Illinois No. 6 seam, was selected as a reference coal.

The Illinois No. 6 seam is one of the most heavily mined, and has one of the largest reserves, in the area. The 1967 annual production was estimated at about 50 million tons, with 62.6 billion tons of recoverable reserves. The coal has a rather high sulfur content, 3.8%, primarily in the form of organic sulfur. Its heating value and fixed carbon, volatile matter, ash, and moisture contents are typical of coals in the eastern region. An analysis of this coal is given in Table 108.

Residual Oil

The northeastern region of the United States is the principal market for residual fuel oil, with the heaviest usage being in the New York to Philadelphia area. Residual oil use in this area is not limited to utilities but is also used for industrial boilers. The principal sources of residual fuel oils for this market are from the refineries of Venezuela, the Netherlands Antilles, and Trinidad. These sources provide in excess of 90% of the total residual fuel oil consumed in the United States. The average sulfur content of these fuels is about 2.54% as shown below (Ref. 189):

Source of Imported Residual Oil	% of Total Import	Average Sulfur %
Venezuela	51.2	2.65
Netherlands Antilles	35.1	2.45
Trinidad	9.7	1.95
Other	4.0	3.35
TOTAL	100.0	2.54

The total United States consumption of residual fuel oil for electric power generation was only 9.1% of the total fossil fuel utilized here in 1966. The consumption of residual fuel oil for all purposes in the United States had remained constant at a level of about 550 million barrels per year for the past 20 years, although it has shown an upward trend since 1965 (Ref. 190). Conversely, the production of residual oil in the United States has been decreasing, and the percentage of residual oil produced from refinery feedstocks has also steadily decreased. This percentage was reported to be 24.4% in 1960 (Ref. 191), 8.1% in 1965, and is projected to be 5% in 1975 (Ref. 192). However, while residual production in the US has decreased, the use of residual oil as a utility fuel has increased. In 1964 about 17% of the residual in the US was used by utilities, but by 1966 nearly 23% of the residual used was for power generation. The figures on an absolute basis are 98 million barrels in 1964 and 141 million barrels in 1966, an increase of about 45%.

The production of residual from domestic crude oils is limited by higher API gravity (lower density), type of refining, and by the demand in the United States for such higher petroleum products as gasoline, diesel fuel, jet fuel, and home heating oils. Caribbean crudes, however, have a lower API gravity (higher density) typically 18 to 25 deg API, as opposed to the average United States crude runs of about 33 deg API. This factor, and also the differing demand pattern, account for the high percentage of residual oil production in the Caribbean. While Venezuelan area crude runs are very much smaller than those of the United States, there is not much difference in production of residual fuel oils. For example, there is approximately 2.5 million barrels per stream day (bpsd) capacity in Colombia, the Netherlands Antilles, Trinidad, and Venezuela which produce some 1.4 million bpsd of residual oils, or about 57% of the crude. At the same time, the United States, with daily crude runs of some 12 million bpsd, produces only 900,000 bpsd of residual oil, or about 7%. In addition, the residuals produced in the United States are widely dispersed geographically and are mainly consumed on a local or quasi-local basis. Hence, the residual oil selected for desulfurization or conversion would probably be imported to the US.

For historic and practical reasons, a hemispheric fuel oil policy has been assumed. Such a policy precludes dependence on crude oil supplies from outside the western hemisphere, i.e., supplies of crude oils from Africa, or the Near East cannot be considered to be reliable, firm, and safe. Nor, currently, can the North Slope or possible Canadian Arctic oils be considered. While these oils are prospectively low in sulfur, it is assumed that they will be more easily delivered to the Chicago area for refining. Presently, no adequate method of transportation has been developed for delivery to east coast refineries.

For these reasons, a Venezuelan crude has been selected as the basis for the residual fuel oil for this study. The detailed specifications and properties for this residual fuel oil are presented in Table 109.

APPENDIX D

COST ANALYSES OF POWER AND FUEL PROCESSING SYSTEMS

POWER SYSTEMS

Cost estimates for power plant components and accessory equipment are difficult to make, especially when similar equipment is not currently being manufactured or where construction materials differ significantly from present practice. However, if the preliminary cost estimates for those power systems having performance levels warranting further consideration are made on a consistent basis, equitable comparisons among different systems can be made. If a preliminary economic analysis indicates that a particular power system is potentially competitive with the base case, i.e., the untreated coal-fired or oil-fired conventional steam station, a more refined cost analysis would then be made.

Method of Analysis

The general method of estimating the capital costs and the annual owning and operating costs of the power station is discussed below. The cost estimates for all three generations of power systems were based on a constant 1969-70 dollar.

Capital Cost

The capital cost of a power station is divided into a number of separate accounts by the Federal Power Commission. Although the exact number of the account is a function of a particular power plant type, e.g., steam station (31X) or gas turbine station (34X), these accounts generally encompass the following:

Federal Power Commission Acct. No.	Description
310 or 340	Land and Land Rights
311 or 341	Structures and Improvements
343	Prime Movers (gas turbine)
344	Electric Generators (for gas turbine)
312 or 342	Steam Generator Equipment

Federal Power Commission Acct. No.	Description
314	Steam Turbine-Generator Equipment
315 or 345	Accessory Electrical Equipment
316 or 346	Miscellaneous Powerplant Equipment
353	Station Equipment
	Other Expenses

For all power systems considered, Acct. No. 310 (340), Land and Land Rights, was assumed to be identical. This assumption introduces little error since this item is small when compared with the other station costs.

The costs of Structures and Improvements, Acct. No. 311 (341), are functions of the number and size of the prime movers and the heat exchangers. These costs would be lowest in power stations incorporating simple-cycle gas turbines since these require no massive heat exchangers, and powerplant foundations are relatively simple. For power stations incorporating fired boilers or fluid heaters, the magnitude of this account will approach that for a conventional steam station. Approximately 70% of this account is in the cost of the actual powerhouse, whereas the remainder comprises costs which are only weak functions of the size of the power system.

The capital cost of the power system prime movers is covered in Acct. No. 343. For those systems utilizing either open- or closed-cycle gas turbines, the cost analysis program detailed in Appendix G was used to estimate machinery cost. Where applicable, the cost of regenerators, intercoolers, precoolers, cooling towers, or other unfired heat exchangers associated with the turbomachinery also were included in this account. Cost estimates for this type of equipment were based on the duty cycle of the heat exchanger and the anticipated operating pressures and temperatures. For gas turbine stations, the cost of exhaust stacks was included in Acct. No. 343. Extensive use was made of information from Ref. 193 in the preliminary estimates.

Account No. 344 covers the costs of the electrical generators for those stations using working fluids other than steam. Generator costs were obtained both from manufacturers' catalogues and from direct quotes.

Fired and unfired steam generating equipment is covered in Acct.No. 312. Preliminary cost estimates for this equipment are based on either the actual costs of similar equipment or estimates submitted by manufacturers. The equivalent of this account for nonsteam systems is Acct. No. 342. Once again, equipment costs were based on the duty cycle and the operating conditions using information from Ref. 193. Account No. 312 (342) includes such items as draft systems, stacks, piping, controls, and other auxiliary equipment associated with the boiler (heater). Approximately 50% of the total in Acct. No. 312 is directly attributable to the cost of the steam generator or the fluid heater. whereas the remainder comprises auxiliary item costs.

Account No. 314 includes costs of steam-driven turbogenerator equipment. Manfuacturers' catalogues and direct quotes were the sources of costs for this equipment. Also included in this account is the cost of the cooling towers, an item which was included in Acct. No. 343 for nonsteam power systems.

Since all power stations require a large amount of electrical equipment not directly involved in the production of power, the cost of this equipment is listed under Acct. No. 315 (345). All cost estimates for this equipment were based on the costs of similar equipment being installed in present-day powerplants.

Miscellaneous Powerplant Equipment, Acct. No. 316 (346), includes costs for tools, shops, emergency equipment, etc., and normally is a function of powerplant complexity. The magnitude of this account is small compared with the other accounts.

Account No. 353, Station Equipment, covers the cost of power transformers, fire-fighting equipment, firewalls, etc. Costs in this account are functions of the number of prime movers and powerhouse size.

Startup expenses, temporary buildings, and other one-time charges are covered under Other Expenses. These expenses were assumed constant for all power systems.

The sum of the individual FPC accounts is the total of the direct costs for a power system. This sum is not the installed cost since a number of other factors also must be taken into consideration in order to arrive at the total installed cost. These latter factors include charges for engineering, design, construction supervision, and contingency (approximately 12% of the direct cost); escalation charges (at 3.5% per annum); and interest (at 7% per annum). The total charges for the last two items are functions of the time needed to construct the power station. The construction time was assumed to be 30 months for gas turbine systems, 36 months for COGAS systems, and 48 months for systems requiring complex boilers or heaters.

Annual Owning and Operating Costs

The annual owning and operating costs for a power station are expressed in terms of mills/kwhr, rather than dollars/yr, and are equivalent to the busbar power cost. In order to determine the busbar power cost, the annual hours of operation must be defined. In all cost estimating it has been assumed that a 70% annual load factor, or 6132 hr/yr of operation, would apply.

The busbar power cost comprises three items: the capital charge, the operating and maintenance charge, and the fuel charge. The capital charge is the yearly owning cost, based on a factor of 14% per annum which includes allowances for interest, amortization, depreciation, insurance, and taxes. The operating and maintenance cost is a function of the power system and is determined separately for each system.

For the purpose of the preliminary analysis and to compare the various power systems, the busbar power cost was determined as a function of fuel cost, i.e., the busbar cost was presented with fuel cost as a parameter as in Figs. 26, 27, 28 , etc. In this manner, the busbar costs could be compared over the anticipated range of fuel costs, i.e., 30¢ to 50¢/million Btu, and thereby the effects of power system capital cost and power system efficiency, or busbar costs, could be determined.

Where necessary, more detailed information on particular cost estimating procedures or on accounting methods is given in the appropriate sections of the text.

FUEL PROCESSING SYSTEMS

Capital Costs

Capital cost estimates were made by obtaining costs for major pieces of equipment from vendors, from standard compilations, and from previous estimates, and then by correcting these costs with the appropriate factors, discussed below, to arrive at a total installed cost. Inflation was taken into account by basing all costs on an Engineering News Record Construction Cost Index (ENR Index) of 1270. The ENR index was chosen because it is more sensitive to the effects of inflation than are other commonly used indices. At the end of 1969 the ENR index stood at 1270.

Correction Factors

The following factors were used to correct the purchased equipment costs in order to determine total installed costs. The commodities cost, i.e., piping, installation, structures, etc., was determined by the procedure given in Ref.190 and was found to be approximately 144% of the purchased equipment cost. The sum of the commodities cost and the purchased equipment cost then gives the total direct cost. The indirect cost, including engineering, was estimated at 35% of the total direct cost (Ref. 194). Finally, a contingency factor equal to 10% of the sum of the total direct cost and the indirect cost was added to arrive at the total instlled cost. An example of these calculations is shown below for a piece of equipment postulated to cost $1000.

Equipment Purchased		$1000
Commodities		
Excavation @ 6.4% equipment	$ 64	
Concrete @ 14.5% equipment	145	
Structural Steel @ 13.1% equipment	131	
Buildings @ 5% equipment	50	
Piping @ 40% equipment	400	
Electrical @ 32% equipment	320	
Instruments @ 25% equipment	250	
Insulation @ 6% equipment	60	
Painting @ 2% equipment	20	
Subtotal		1440
Total Direct Cost		2440
Indirect Costs @ 35% of Total Direct Cost		854
Subtotal		3294
Contingency @ 10% of Total Direct-plus-Indirect Costs		329
Total Installed Costs		$3623

Equipment Costs

Purchased costs for general process equipment such as pumps and heat exchangers were taken from standard compilations of costs for the chemical process industries (Refs. 195 and 196). Costs for towers and other cylindrical vessels were estimated by calculating the weight of the vessel (Ref. 197) and multiplying by a fabricated cost for steel construction of 45¢/lb. Where an internal insulation or a refractory lining was required, the cost of this item was added to that of the steel structure (Ref. 195). Final estimated cost of this process equipment was made by using a Lange factor of 3.7.

The cost estimates for large centrifugal compressors were based on data from Ref. 198; where costs for compressor sizes other than those noted in this reference were required, the new costs were determined by scaling the airflow rating by an 0.8 exponent. For example, to estimate the cost of a compressor with a rating twice as large as the one for which cost data was available, the cost of the base compressor would be multiplied by $2^{0.8}$, or 1.74 (Ref. 195).

Cost estimates for the Lurgi Gasification System was based on information from Refs. 199 and 200. These estimates were increased by 10% to account for interfacing the Lurgi process with the remainder of the plant. Costs were estimated at $19/kw for a first-generation plant, and $15/kw for a second-generation plant.

Costs for the Wellman-Galusha gasifier were based upon information from Ref. 201. The cost of multiple units was estimated by multiplying the unit cost by the number of units raised to the 0.9 power (Ref. 202). Thus, 100 units would cost $100^{0.9}$, or 63 times the cost of a single unit.

Costs for the COED pyrolysis process (Refs. 203, 204, and 205), the Kellogg Molten Salt Gasification Process (Ref. 206) and the Consolidation Coal "GASOLINE" Process (Ref. 207) were obtained from previously published estimates for these processes. Process plants were scaled to the size required for a 1000-Mw, advanced power cycle by a 0.7 exponent. Thus, the cost of a plant twice as large as the one estimated previously would be obtained by multiplying by $2^{0.7}$, or 1.625 (Ref. 196).

The above methods were used to prepare detailed cost estimates for those fuel processes which could meet the required sulfur oxide emission goals. These cost estimates are given in Table 110 for the first-generation Lurgi-type gasification system and in Table 111 for a second-generation Texaco-type gasification system. Table 112 gives the costs for alternative equipment for use in an iron oxide sulfur removal system suitable for use with the second-generation gasification system, Table 113 gives the estimated costs for a third-generation Texaco-type gasification system, and Table 114 gives the costs for alternative equipment for use in a dolomite acceptor sulfur removal system for use with the third-generation gasification process.

Annual Owning and Operating Costs

Capital costs are presented on a dollars-per-kilowatt basis and are based on the net electrical output of the total plant, i.e., 1000 Mw from the power cycle minus the electrical requirements of the fuel processing section.

The estimates of annual operating costs postulate the supply of a clean fuel to a nominal 1000-Mw system with an annual load factor of 70%. It was assumed that the utility would own the fuel processing plant and that utility-type financing would be applicable.

As noted, annual capital charges were estimated to be 14% of the total fixed capital charges including equipment and general facilities. Annual maintenance costs were estimated at 5% of total fixed capital, and although this value may be somewhat conservative, it reflects the long lifetime of the fuel treatment plant.

Annual labor costs were estimated at $15,000/man/shift based on 4 shifts of labor per day and operating labor at $3.65/hr. This charge includes supervision at 20% of labor, payroll added costs (PAC) at 10% of labor-plus-supervision, and overhead costs at 50% of labor-plus-supervision-plus-PAC.

Utility costs are standard items for chemical process industries. They are: cooling water at 2¢/thousand gal, process water at 30¢/thousand gal, and steam at 50¢/thousand lb. These costs include the cost of water, water treatment, fuel, and power.

The costs of the more common raw materials and the credits taken for by-products are given in Table 115. By-product sulfur from the processes was credited at $25/long ton, which has been an average value for the price of sulfur over the last 10 years. The effect of sulfur credit on annual operating costs for the fuel processing part of an integrated power station is slight. An incremental price change of $5.00/long ton for sulfur is equivalent to only about 0.04 to 0.08 mills/kwhr, depending on the amount of sulfur removed and the efficiency of the power generation cycle.

APPENDIX E

HIGH-TEMPERATURE DUST REMOVAL

It is not the intent of the following discussion to provide a full discourse on the theory and practice of dust collection, but some background information is appropriate as an introduction to the process design that will follow. The types of collectors discussed include filters, cyclones, electrostatic precipitators, mechanical collectors, and rotary stream dust collectors. Wet scrubbers, although highly efficient, are obviously unsuitable for high-temperature dust removal. Typical characteristics of several types of dust collectors are shown in Table 116 (Ref. 208).

Characteristics of Dust Collectors

Filters

Filters operate by allowing the gas to pass through a porous structure which retains the particulate matter. Removal efficiencies can be high for small particles, and high-temperature gas can be handled with the proper materials of construction. Large surface areas may be required to achieve high efficiency, and provisions must be made for cleaning collected particles from the filter material. Expendable filter material may be used, in which case provisions must be made to replace the filter material.

High-velocity filters operate, in principle, more like inertial separators than filters. High-velocity gas passes through a porous filter where particles are removed by impinging to and sticking to an obstruction in the flow path, since they will not change direction with the gas. Very small particles can be removed at high gas velocities.

Cyclones

Cyclones are devices in which organized vortex motion created within the collector provides the force to cause particles to be concentrated in certain areas from which they are removed. Since the forces vary inversly with the radius of the cyclone, small cyclones are more efficient than large cyclones and can remove smaller particles. However, the capacity is less for small cyclones, so that for high-efficiency removal of small particles, multiple small cyclones are used in a parallel arrangement. Operating temperature is limited by materials of construction.

Electrostatic Precipitators

Electrostatic precipitators use a high-intensity electrical field to charge the particles to be collected. The charged particles are then acted on by electrical forces and are deposited on a charged collecting surface. Since the charge is applied only to the particles, pressure drop is low. Collection efficiency varies with residence time, which may be large for collecting small particles. For high-efficiency operation, precipitators may be placed in series. In this type, also, operating temperature is limited by materials of construction.

Mechanical Collectors

Mechanical collectors include a wide variety of devices that separate particles from gases by mechanical forces. These devices include settling chambers, baffled chambers, or various types of inertial collectors using centrifugal forces. Capacities and efficiencies vary widely depending upon the operating principle of the device and its application.

Rotary Stream Dust Collectors

For the application to a high-temperature gas stream (to 1500 F), a typical design would be based on cleaning the gas to a particular concentration which could be handled by a gas turbine (nominal target of about 5 ppm, nothing larger than 5 microns).

The hot gas to be cleaned would come from the dolomite acceptor process, for example, where it is assumed to be at 1500 F, and would have a dust loading of about 8000 to 9000 ppm. This dust would come from ash originally in the coal (or from residual oil as well) and from undersized dolomite particles that become entrained in the gas. It is further assumed that 50% of the particulate matter is less than 5 microns.

The rotary stream dust separator is similar to a cyclone in that it uses, primarily, centrifugal forces in a confined vortex, and is without moving parts. It may be defined as a cylindrical flow tube in which the main gas stream forms an inner vortex, opposed by a secondary outer vortex flowing in the same tangential direction but in the opposite axial direction, so that the combined centrifugal forces and radial gas flows concentrate the dust particles in an annular ring.

A typical device is shown in Fig. 153. The bulk of the description and performance data come from Klein (Refs. 209, 210, 211, and 212).

Mechanism of Operation

The entering dust-filled gas passes through the annular space between the dobbas and the inside walls of the inlet port. The inside diameter of the inlet port increases slightly toward the open end, and the presence of the dobbas stabilizes the flow pattern and aids in establishing the inner vortex at a lower point in the tube. The outer vortex caused by the secondary air jets creates a vortex pattern of flow in the entering gas.

The secondary air jets, entering at a tangential angle near the top of the tube, create an outer vortex (potential flow) with a downward axial component and an inward radial component. This potential flow creates an inner vortex (rotational flow) in the same direction of rotation but with an upward axial component. The result of these forces is the concentration of the dust in an annular space, called the "mixed flow" region. This dust-rich layer is conducted downward, outside the lips of the inlet port, and into the dust bunker below (or withdrawn separately) as part of a purge flow, with a compensating return flow from the bunker to the dirty gas inlet.

The baffle ring at the top serves to block the portion of the uppermost jet flow that would otherwise escape from the top of the tube, and markedly improves the efficiency.

Performance

Typical dimensions of an 8-in. unit are shown in Fig. 154. Fractional efficienices are stated to be 100% for 5-micron particles and 90% for 1-micron particles having a density of 2.65 g/cm^3.

The efficiency variation with dust load is negligible over the range of 1 to 200 g/m^3 (0.4 - 88 g/ft^3). This wide range of high efficiency is attributed to the fact that the dust is introduced near the zone of separation, and is separated in a space filled with well-defined and controllable flow patterns, without deleterious eddy currents.

The efficiency characteristics of 8-in. and 20-in. diameter tubes are equal, but larger sizes (40-in. diameter) show a decreased efficiency. On fine-grain coal dust, 98% collection was obtained with 8-in. and 20-in. units, and 97% with a 40-in. tube. The 8-in. diameter tube has a capacity of 300 ft^3/min of gas.

In terms of efficiency, the rotary stream dust separator appears to be definitely superior to conventional cyclones, and at comparable total power costs. The efficiency is in the range of the better wet scrubbers and ordinary applications of dry electrostatic precipitators. In spite of the fact that increasing tube diameter is less harmful to efficiency than it is with cyclones, it is still deleterious; thus, the highest efficiency must be obtained with multiple-bank installations of small tubes. These, as with banks of small cyclones, must be designed for reasonably uniform distribution of pressure. The effect of dust load distribution is less significant than for normal cyclone separators.

Another important advantage is the absence of erosion problems on the tube wall, and the absence of a need for smooth internal construction of the tube. Minor turbulence caused by welding beads, etc., has no detrimental effect.

On the other hand, the rotary stream separator does require a source of higher-pressure secondary air, and presumably, a high-pressure blower. Furthermore, for optimum performance, the secondary air pressure and flow rate must be controlled in relation to the flow of primary gas; this could be an advantage, because by exercising such control, efficiency may be kept at a peak over a wide range of operating conditions.

For the present application, the requirement for a source of secondary high-pressure gas is easily met. It is assumed that combustion air is used and premixed with the fuel. Since the combustion air has to be compressed anyway, a small amount may be added at the dust removal point without having a significant effect on process costs.

Collection System Design and Cost

For the application under consideration, it was estimated that the gas leaving the rotary stream separator would contain 30 ppm of dust with nothing larger than 5 microns. With a capacity of 300 actual ft^3/min for an 8-in. tube, approximately 440 tubes would be required. For high-temperature operation, stainless steel construction would be indicated. The installed cost was estimated at $1,500,000.

For a final gas cleanup, high-velocity impact filters have been selected to operate with a gas velocity of 600 ft/min, with a filter area requirement of about 220 ft^2. To allow replacement of the expendable filters, parallel units would be used, each of full capacity. These are estimated to cost $350,000 for stainless construction in a pressure frame. The filter medium is expendable and was assumed to be asbestos, or some other high-temperature fiber, in a felt or mat, thereby eliminating the high cost of woven cloth. Filter costs were estimated at $600,000/year with user costs of 10¢/ft^2. Labor, primarily for filter changing is estimated at $300,000/year. Other operating costs are insignificant.

The fuel gas finally delivered to the gas turbine is estimated to contain 3 ppm of particulates with nothing larger than 5 microns.

APPENDIX F

DETERMINATION OF THE PRODUCTS OF COMBUSTION

The determination of the temperature rise following the combustion of a fuel in air is a particular application of the more general problem of determining the thermodynamic properties of a working fluid consisting of a gas mixture of many constituents. Since the gas mixture would be formed at an elevated temperature by the combustion process, it is usually a good approximation to assume that the gas mixture is at thermodynamic equilibrium. The evaluation of the thermodynamic properties of a system of known composition is straightforward. However, the determination of the equilibrium composition of a system of many constituents is a computational problem of considerable difficulty and, therefore, solution of the problem becomes a practical reality only if a high-speed digital computer is employed.

The computational program employed by the United Aircraft Research Laboratories (UARL) for the determination of the equilibrium composition and thermodynamic properties of combustion gases is based on a method originally developed by Brinkley (Refs. 213, 214, and 215). This method has been revised at UARL and adapted for the solution of specific combustion problems involving air-breathing and rocket propulsion devices under study by UARL.

With this method, the equilibrium composition and temperature of a reacting mixture are obtained by applying a successive-approximation procedure to find the simultaneous solution of the standard equations of chemical equilibria, conservation of (atomic) mass, and conservation of energy for specified values of pressure and either temperature or enthalpy. The derivation and solution of these equations involves the following simplifying assumptions: (1) steady-state conditions, which can be defined for the combustion chamber, are assumed to exist, (2) complete thermodynamic equilibrium is obtained for the combustion products, and homogeneous mixing occurs, (3) the gaseous products of reaction behave ideally and follow Dalton's law of partial pressures, and (4) the combustion is adiabatic at constant pressure.

The basic procedure involves a matrix method for the solution of the equations listed above. An initial estimate of the temperature and composition gives an approximate solution to the problem which is refined by expanding the set of equations to be solved in a Taylor series around differences between the true solutions and the trial solutions and dropping all exponents higher than the first power in the Taylor expansion. The system of equations is then solved as a set of linear simultaneous equations by the Newton-Raphson iterative method until the desired accuracy is obtained. The mathematical treatment of the procedure described above has been developed extensively by Brinkley and may be found in Refs. 213, 214, and 215.

The results obtained from the calculation procedure contain not only the temperature and composition of the reaction products, but also the density, enthalpy, entropy, molecular weight, gas constant, specific heat, and the ratio of specific heats.

The above procedure was applied to the combustion of the prospective fuel gases given in Table 117 as well as to the combustion of methane. The results are shown in Figs. 155 through 159 for Fuel Gas I, Figs. 160 through 165 for Fuel Gas II, Figs. 166 through 173 for Fuel Gas III, and Figs. 174 through 177 for methane. Since Fuel Gas I and Fuel Gas II would be produced in conjunction with low-temperature sulfur-removal systems. temperature-rise calculations were made for only low fuel temperatures and fuel temperatures equal to the combustion air temperature. There exists the potential for using Fuel Gas III at much higher temperatures; thus temperature-rise calculations were made for a number of fuel supply temperatures ranging from 80 F to 2200 F.

Fuel gases numbered IV and V were not subjected to the above analyses, as it became apparent that for fuel gases derived from the various partial-oxidation processes, either high- or low-temperature, and burned in an advanced-cycle power system at similar equivalence ratios, similar combustion product properties and temperature rises would be obtained.

It is interesting to note that the effect on power system performance of the different heating values for the low-Btu gases considered is slight. Table 118 shows, for a given power system, that the maximum deviation from the mean system efficiency when burning a low-Btu gas is only 2.4%.

APPENDIX G

DESCRIPTION OF GAS TURBINE COST MODEL

The gas turbine engines being considered as prime movers for advanced-cycle, base-load electric power generating stations which could make their appearance in the period from 1975 to 1995 represent significant advances in the state of the art, both from the standpoint of operating conditions and that of unit output power. However, it is impossible to predict accurately the impact these engines would have on the future of electric power generation without an accompanying economic analysis since these future gas turbine engines would be characterized by entirely new design criteria. By contrast, many present-day industrial gas turbine engines are merely adaptations of aircraft engines in which light weight and high specific power (lb thrust/lb airflow) were emphasized. As a result, few of the general economic correlations developed for evaluating present-day industrial designs could be expected to hold true for future designs, primarily because many of the required geometric and economic scaling factors are not well defined. Further, these future industrial-type gas turbine designs would make use of innovations not applicable to current machinery in order to attain the performance necessary to be competitive with alternative methods of generating base-load electric power. The significance of these observations is that, except for the design of aerodynamic flow passages (primarily the vanes and blades), and use of aircraft-type advanced materials, future large-scale engines will be designed and manufactured with much different philosophies than are considered common today.

Therefore, it was necessary to develop an entirely new analytical procedure which would estimate accurately the economics of advanced gas turbine engines capable of producing large amounts of power in base-load central power stations.

Economic Analysis

Descriptions presented in this section are concerned primarily with the economic analysis which estimates costs and selling prices of future gas turbine engines incorporating all standard accessories such as the fuel system, inlet housing, and exhaust stack. Cost estimates for peripheral equipment such as pumps, generators, housings, mountings, etc. are not included here but are taken into account in the overall system costs.

A survey was made of present-day gas turbine engines manufactured by Pratt & Whitney Aircraft, General Electric, and others to determine whether any correlations between price and engine characteristics existed, and if so, whether these could be used in a gas turbine model synthesized to reflect technological developments anticipated for the forthcoming years. Unfortunately, no general correlations of this type were found to exist, primarily because most of the current engine models were designed for entirely different types of service where, as noted, differing sets of design criteria ultimately dictate final configurations. As a result, it was found necessary to establish a model for the economic analysis which synthesized an engine design and summed the costs of the various major components in such a manner that overall costs could be developed in a building-block fashion. These costs were obtained from the literature where possible, from vendors' estimates, and from qualified UA Division personnel.

It is interesting to note that practically every source of economic data indicated that the primary variables affecting gas turbine engine component costs were: (1) the component's geometry (primarily its linear dimensions); (2) the material selected; and (3) the production volume. Other variables were estimated to have little effect on cost estimates in a generalized economic program suitable for analyzing many different gas turbine engine designs.

A schematic diagram of the model developed to estimate manufacturing costs of advanced gas turbine engines is shown in Fig. 178. Detailed cost relationships were developed for the low and high compressors, the burner, the high, low, and power turbines, the shafts, the bearings, and the casing and exhaust elbow. The costs of the inlet housing, fuel system, and miscellaneous small parts were lumped together and considered to comprise a constant fraction of the overall manufacturing cost. Provisions in the economic program were made to analyze engine designs incorporating compressors with either constant hub, mean, or tip diameters. Different blade and disk materials (see Table 119) can be specified for each compressor and, in addition, provision is made to accommodate a change in blade and disk materials in the high-pressure compressor. Since the computer design and performance program (Part 2, Section II) does not specify the number of blades and vanes in the compressor section(s), the economic analysis includes provisions to calculate the number of blades and vanes for each stage, based on acceptable engineering practice. A compressor gap-to-chord ratio of 1.0, representative of that for advanced engines, and an average blade stagger angle of 30 degrees (relative to the engine axis) were selected. The number of blades then was calculated from the hub diameter and the blade length-to-chord ratio.

The burner was sized using volume flow, reference velocity, and length-to-diameter (or height) relationships, and either an annular (indicative of advanced engines) or a cannular (indicative of present-generation engines) design can be accommodated. In those cases where exotic working fluids such as carbon dioxide, sulfur dioxide, or helium were studied, the burner calculations were omitted. In these circumstances, fixed heat exchangers would be used instead of a burner, and the calculations for these are considered separately. Each stage of the turbine section was analyzed individually as were the compressor stages. Varying hub and tip diameters as well as differing blade and disk materials (again see Table 119) can be accommodated. Provisions also have been made to handle impingement cooling in the turbine blades and the use of tip shrouds where necessary.

As many as three shafts (two in the gas generator section and one in the power turbine) can be included in the overall analysis. The shaft diameters were calculated by using material stress relationships and rotor torque. Casings for each section of the engine were assumed to be cast in halves, and final machining was assumed to be conducted in the engine manufacturer's facilities. Casing thickness was based on either the minimum thickness required to withstand internal gas pressures, or the minimum thickness required for sound casting techniques. In the former case, a metal working stress of 30,000 psi and a factor of safety of 5 have been assumed. The exhaust elbow was considered to be made from sheet steel and be similar in design and fabrication to those elbows used on present-day engines. A choice of bearings includes sleeve, Kingsbury thrust, roller, and ball types, the prices of which would vary primarily as the bore diameter of the type selected. In those engine designs not requiring bearings at a particular location, provisions have been made to omit that particular component cost estimate. As mentioned, several specific engine components (fuel handling equipment, inlet housing, nuts, bolts, etc.) were combined, and the associated cost of these parts was assumed to be equal to 18% of the total manufacturing costs.

Since most of the input information required for this anlaysis was based on dimensions of the components, only a slight change in input data would be necessary for engines using working fluids other than air. Every effort was made to make this economic analysis as detailed and as versatile as possible and, although nearly 90 different pieces of input data must be supplied for each case, most of these data are obtained directly from the thermodynamic design program. This versatility makes it possible to increase the reliability of the results in support of the technical analyses in this program.

Component Cost Information

Once the basic economic model was established, it was necessary to consult with vendors* to obtain costs for the different major parts within the engines. Each vendor contacted was supplied with illustrative drawings and a list of materials which were thought typical of the part to be manufactured and then was asked to supply price information in 1969-70 dollars. Where necessary, tables of nominal dimensions were also supplied to the vendors in order to provide them with additional information which would cover a range of parts sizes to be considered in this study.

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* Vendors are identified by number (No. 1, No. 2, etc.) in accordance with an identifying code list provided, separately, to NAPCA.

Compressor Section

Drawings of solid compressor blades with aerodynamic blade lengths from 6 in. to 36 in., similar to those shown in Fig. 179, were supplied to representatives of blade manufacturing organizations. In return, these vendors (No. 1 and No. 2) supplied price estimates from which Fig. 180 was constructed. These prices, primarily noted as ranges in this figure, are for blades with straight pedestals, (i.e., a constant hub diameter design) manufactured from AMS 4928 material. The vendors noted that mid-span rod stiffeners would be required on blades with lengths greater than 24 in. and length-to-chord ratios greater than 3.0. These stiffeners are quite expensive if cast integrally with the blade, but for the industrial designs considered, the representatives estimated that round rods inserted through, and welded to, each blade would increase the cost of each blade by only about $2.00. Sloping pedestal blade designs would be approximately 15% more expensive than the prices noted in Fig. 180, primarily because of the increased machining required on the pedestal. It was recommended that changes in blade materials could be accommodated by assuming that 40% of the blade price would be material cost and 60% labor charge. Different blade materials were handled in the analytical program by taking the overall blade dimensions of length, height, and width, calculating the volume of this block of material, estimating the price of this block if made from the "new" material, then replacing the original material fraction with the new estimate in the basic blade price analysis. Tooling costs for each blade design were estimated at $28,000, a value which was assumed to be written off over a five-year production run. Individual compressor vanes were assumed to be formed from strip stock at a price of $2.40 per foot. The vanes would be inserted into a vane ring holder, and the cost of material, insertion, and welding was estimated to be $7.10 per inch of rim circumference. Finally, the cost of spacers between adjacent stages was estimated to average $400 per stage.

Drawings for typical compressor disks similar to that shown in Fig. 181 were sent to a manufacturer of some of the largest forgings in the US. A table accompanying this drawing listed representative dimensions of disks with diameters from 3 ft to 11 ft as manufactured from AMS 6304 and AMS 5661 materials. Representatives from this vendor (No. 3), in turn, supplied the price estimates shown in Fig. 182. These estimates were based on production of at least 50 disks per year in one run. The estimates would increase by 25% for a run of 20 disks, but would decrease linearly to the values shown in Fig. 183 for production rates between

20 and 50 disks. Tooling costs for compressor disks with diameters less than 8 ft are shown in Fig. 183 for disks produced from both AMS 6304 and AMS 5661 materials. Above a diameter of 8 ft, open dies would be necessary and these would be maintained by the manufacturer at no cost to the purchaser. It was recommended that the forged disk prices be increased by 30% to cover finished machining costs.

Engine Burner

The costs of the engine burners were estimated to be proportional to the surface area of the burners. Surface area, in turn, was found from the volume of the burner, the design reference velocity, and the length-to-diameter (or length-to-annular height) of the burner. Combustion chambers were estimated to cost \$400 per ft^2 and \$600 per ft^2 for cannular and annular burner designs, respectively. Accessories and manifolds would add an estimated \$35 per lb of airflow to this cost. Provisions were included in the analytical program to maintain the maximum burner diameter within certain present limits, and adjustments in burner length were made if this overall diameter was exceeded.

Turbine Section

It was noted in the discussion of prices for compressor blades that a technique was available to estimate prices of solid (noncooled) blades produced from materials which differ from that for which estimates already had been supplied. As a result, it was not necessary to obtain separate price estimates for solid turbine blades from the manufacturers. However, impingement-cooled turbine blades (and vanes) require entirely different manufacturing techniques from those used to produce solid blades and, consequently, separate price estimates were needed. Therefore, blade and vane drawings similar to those shown in Fig. 184 were supplied to several manufacturers. Price estimates received from two of these vendors (No. 2, and No. 4) are presented in Figs. 185 and 186 for blades and vanes, respectively, produced from B-1900A and Inconel 713C materials. From these figures, it can be observed that impingement-cooled turbine blades would be considerably more expensive than solid blades with the same dimensions and materials. Tooling costs for blade and vane designs with airfoil lengths less than 11 in. for blades and 7.5 in. for vanes were estimated to be \$100,000 each, while tooling costs for blade and vane designs beyond these lengths were estimated to be \$150,000 each. All turbine blade tooling costs were assumed to be amortized in five years.

Direct contacts with vendors revealed that, if it were necessary to cast tip shrouds integrally with turbine blades, the price estimates presented both in Fig. 180 for solid blades and in Fig. 185 for hollow blades must be increased by 10% to allow for additional casting complexity and machining. Machining the root section of blades for engine designs with varying hub diameters would add an additional 15% to the turbine blade prices. No estimates were given for the variation of direct manufacturing costs with production volume, but the vendors remarked that the volume of blades and vanes produced should be sufficiently high to preclude any upward adjustments in the prices per unit.

Turbine disk drawings, similar to those shown in Fig. 187 were supplied to Vendor No. 3, and their price estimates are presented in Fig. 188 for disks with diameters ranging from 3 ft to 11 ft manufactured from two materials, AMS 6304 and AMS 5661. Since most of the dimensions on gas turbine disks would be essentially proportional to disk diameter, irrespective of the absolute value of disk diameter, the fact that the prices of both the compressor and turbine disks would be roughly proportional to the third power of the diameter (or,essentially, to the disk volume) is not surprising. Representatives from Vendor No. 3 noted that the tooling costs for turbine disks would be the same as those for compressor disks, varying only with disk material, as shown in Fig. 183. It was stated that, if disks with diameters larger than 11 ft were foreseen, entirely new forging facilities might be necessary since no commercial facility exists, other than that presently operated by the US Air Force, which would be capable of producing such large parts. The cost of such a new facility could be in the several-million-dollar range, all of which would have to be charged to the production of engine disks should no further commercial applications be developed.

Engine Bearings

Representatives from Vendors No. 5 and No. 6 were contacted to obtain price estimates for large, antifriction, roller and ball bearings. These representatives were quite cooperative in supplying price estimates for bearings with differing bore diameters, and correlations of these data are presented in Fig. 181. All data are for oil-damped bearings manufactured from M-50 carburized steel in lots of at least 50 bearings per year. If lots of less than 50 bearings were made, the unit prices would increase by approximately 50%. Price estimates from personnel indicate that purchase prices for sleeve bearings and Kingsbury thrust bearings would be approximately 40% and 50%, respectively, of those estimates for roller bearings with the same bore dimensions. To each bearing price must be added a charge for bearing supports, estimated to be approximately 50% of the price of the bearing alone. Bearing locations throughout the analytical engine model, the number of bearings specified for each location, the shaft on which the bearing rides, and the bearing option open to the engine designer are shown in Fig. 190.

Shafts and Casings

The price of turbine engine shafts was estimated to be $10/lb based on data from Ref. 216. Shaft weights were calculated from an estimate of shaft torque, shaft length (a function of engine turbine and compressor stages and burner length), shaft diameter (a function of engine power rating), and the material density.

Consultations with UA Division engineers revealed that, based on recent experience, casings would cost approximately $1.50/lb. A charge equal to 30% of the raw casting cost was added for setup and in-house machining. As noted previously, casing thickness would be dictated either by the internal gas working pressure or the minimum thickness necessary to obtain sound castings. The cost of the exhaust elbow was estimated to be $10,000 for an engine with a throughflow of 250 lb/sec, and proportional to the square root of the airflow for larger engines.

Assembly Costs

Assembly and test costs are difficult to assess accurately. Based on recent industry experience, these costs should vary between $75,000 and $150,000 per engine, depending on engine complexity and size. In circumstances where complete engines would be too large to ship to the erection site in a completely assembled package, it was assumed that subassemblies would be shipped to the site and assembled in position. Whether the engine were shop-assembled or field-erected, the cost of assembly is included in the manufacturing cost as if all units were shop-assembled. X-ray costs might be as high as $15,000 per engine inspected, and when extremely large units are built, or units using working fluids other than air are assembled, it would be necessary to X-ray every unit assembled. For smaller, open-cycle engines, X-ray inspections would not be required as frequently, and consequently, the X-ray cost per engine produced would be less.

Selling Price

The summation of individual engine component costs, assembly and test charges, and X-ray costs, comprise the manufacturing cost per engine. In the analytical program, it was assumed that many of the component parts would be supplied in finished form by the vendors and, where not, appropriate additional machining charges (including overhead) were added. Manufacturing overhead charges were assumed to be similar to those of industrial manufacturing plants, and not to those which have been found necessary in high-technology organizations.

Although it would be essential that a manufacturer know his production costs, the ultimate electric utility purchaser would be concerned primarily with the final purchase price he must pay for a piece of machinery. Therefore, in order to develop a selling price which must be charged by an engine manufacturer, additional economic data must be specified. These include: the total development cost of the engine (including additions to existing manufacturing facilities needed for engine production); the average engineering expenses needed to support the production and operation of a particular engine design; the gross profit per unit produced; and the general and administrative (G&A) expenses allocable to the particular program. In addition, an estimate of market penetration must be made to schedule production volume and to estimate amortization of tooling costs during production. By taking the simplified approach of allocating development, continuing engineering, profit, and G&A among all engines produced, the selling price was developed directly from manufacturing costs. In this analysis, expenses such as sales and field engineering support are treated as separate costs, and, in all likelihood, the manufacturer would increase the selling price per unit to cover these expenses. The analytical program which was developed in support of the technical analyses of this study does not include an allowance for these latter charges; rather, they are accounted for separately. Manufacturing costs and selling prices per kilowatt were calculated simply by dividing the respective overall costs and prices by the net power output rating of the engine expressed in kilowatts.

Typical Examples

Results obtained from this analytical cost program which are typical of those utilized in the study are presented in Tables 120 and 121. In Table 120, the manufacturing costs are presented for a 150-Mw, two-shaft, second-generation engine design with a turbine inlet temperature of 2800 F and an overall pressure ratio of 32. This engine would have cooled high and low turbine blades, and shrouds on all low-pressure turbine and power turbine blades. It can be seen in Table 120 that the component cost breakdown is quite extensive, thereby allowing the designer to assess quickly the effects of overall engine cost of such changes as material selection, compression stage split, and burner design. In Table 121 the number of blades and vanes per compressor stage, and the component and overall lengths, are specified. It is from such an analysis, coupled with results from the computer performance and design program, that the drawing of a typical engine (Fig. 78) was produced.

Manufacturing cost and selling price estimates for complete gas turbine engines with rated power-generating capacities ranging between about 50 Mw and 350 Mw are shown in Fig. 191. These estimates are for second-generation engines of the type described in Tables 120 and 121. The estimated manufacturing cost and selling price per kilowatt decrease with output power rating as one would expect with economics of scale. The cost advantages of using larger-size units will, however, be greater than implied by the modest slope of Fig. 166 when total costs for a completed power station are considered. For a given size of plant (nominally 1000 Mw in this study), the use of smaller units would increase the costs of piping, require a larger flow area, result in higher set-up costs, etc.

Concluding Remarks

An analytical program was developed that allows the estimated manufacturing cost and selling price of large industrial gas turbines to be determined with reasonable confidence. Although this program was established on a very broad base to allow an engine designer a wide latitude in selection of materials and design options, it is felt that the accuracy of such an analysis is only as good as the values from which it was developed. Every attempt was made to obtain vendor prices and knowledgeable estimates that were as accurate as possible based on the time period and technology required. Therefore, the analytical procedure was applied to existing production gas turbine units to check its validity. The results indicated that the estimates were within 11% of quoted values. Consequently, a high degree of confidence can be placed on the results of the program which forms a very important part of the overall costing procedure used to estimate the busbar cost of electric power generated by advanced-cycle power systems.

APPENDIX H

THERMODYNAMIC AND TRANSPORT PROPERTIES OF SULFUR DIOXIDE

In order to evaluate the technical and economic feasibility of base-load power-generation systems using sulfur dioxide as the working fluid, parametric analyses of power cycle performance and system sizing must be undertaken. To perform the required cycle analysis, the thermodynamic properties of sulfur dioxide, e.g., enthalpy, entropy, and specific volume, must be known for any state point which lies within the operating pressure and temperature ranges of the cycles of interest. These properties are normally presented in the form of a working chart such as a Mollier chart or temperature-entropy diagram. For the present study, the latter is favored because of its convenience in the construction, comparison, selection, and analysis of different power cycles. For the type of power-generation systems considered, the temperature of the system ranged from 100 F in the condenser to 2200 F at the turbine inlet, and pressure varied from 84 psia in the condenser to 4500 psia at the turbine inlet.

For the desired temperature range, thermodynamic properties of sulfur dioxide were available only at low pressure, 10 psia to 1000 psia, (Ref. 217). At pressures above 1000 psia, properties were available only from 320 F to 480 F, with complete data in the two-phase region (Refs.218 and 219). Thus, thermodynamic properties in the region between temperatures of 480 F and 2200 F, and between pressures of 1000 psia and 4500 psia had to be determined. These values were calculated using the law of corresponding states to correct for the departure from the ideal gas behavior, Specifically, two types of thermodynamic processes were used to calculate the unknown property of a final state from a known initial state -- a constant-enthalpy process to calculate the change of temperature associated with the increase of pressure from 1000 psia to a given higher pressure, and an isothermal compression process to calculate the change of entropy. The results obtained were combined with the data available in the literature to construct the T-S diagrams, shown in Figs. 192 and 193. The specific volume of sulfur dioxide was calculated using the compressibility factor. Thermodynamic properties of compressed sulfur dioxide liquid were calculated to estimate the work input to the pump.

Transport properties such as specific volume (or density), heat capacity, viscosity, and thermal conductivity were also needed to determine the size of the powerplant equipment, including condensers, regenerators, pumps, compressors, and turbines. The density of the saturated liquid and the saturated vapor in the temperature range 0 to 315.4 F are given in Fig. 194, as is the heat capacity at 1 atm.

The viscosity and thermal conductivity of sulfur dioxide at 1 atm are given in Fig. 195 (from Ref. 220). These properties were not corrected for pressure since they were to be used in the heat transfer calculations in which all properties were evaluated at the surface temperature. At these temperatures, the pressure corrections were negligible.

APPENDIX J

OXIDES OF NITROGEN EMISSIONS FROM GAS TURBINE-TYPE POWER SYSTEMS

Although nitrogen oxide (NO_x) emissions from a power system depend upon many design and operating factors, gas turbine-type power systems usually emit less NO_x (on a ppm stack gas concentration, or pounds per unit heat input basis) than do reciprocating internal combustion engines or large steam power boilers. Reported values of NO_x emissions vary from source to source, obviously because of their dependence upon a large number of factors. Nonetheless, gas turbine-type power system NO_x emissions usually range from 75 to 130 ppm (Ref.221), whereas coal-fired steam boiler NO_x emissions may be as high as 1200 ppm (Ref.222). The NO_x emissions from oil- and gas-fired steam boilers are generally lower than from coal-fired units, and values as low as 150 ppm have been reported in some gas-fired steam boilers (Ref. 223). It should be noted however, that efforts to reduce NO_x emissions from steam boilers have also resulted in reduced operating efficiencies.

The combustion of a fossil fuel with air in all types of power systems results in the formation of nitrogen oxides. Because 90% or more of the NO_x in the stack gases is present as the relatively unreactive nitric oxide (NO), methods to control NO_x emissions by stack gas removal are quite complex (Ref.224) and appear to be unattractive unless they might possibly be combined with sulfur dioxide stack gas removal. Thus, the most practical method of limiting NO_x emissions is to control their formation in the combustion process itself and in the subsequent processes by which the combustion products undergo cooling before being emitted from the stack.

Theory predicts that values of NO_x concentration in the combustion products are dependent on design and operating conditions, flame temperature, excess air (atomic concentrations), and residence time. The formation of NO_x, which begins with the onset of combustion in the primary combustion zone where temperature is a maximum, continues at a relatively slow pace (due to low chemical reaction rates) as colder bulk gas enters the recirculating flame zone. The NO_x formation cannot proceed until high temperatures are achieved through the combustion of the hydrocarbon fuel with air, and therefore, the hydrocarbon chemistry is virtually completed before NO_x formation begins. The amount of NO_x formed thus depends on the conditions in the primary zone and the subsequent temperature and concentration history of the combustion products with time.

It may be noted that since the NO_x is formed after the hydrocarbon chemistry has been completed, the potential exists for controlling NO_x while achieving good combustion efficiency. Although present theory is useful in pointing out the approaches which may be employed to limit NO_x emissions, it is not comprehensive enough to explain all the interactions involved in the chemical, thermodynamic, and fluid dynamic processes. Experience has shown that NO_x emissions may vary extensively between power systems having similar operating conditions, and may even be widely different in identical equipment. Thus, no attempt was made to calculate NO_x emissions in this study using theoretical equilibrium or chemical kinetic procedures.

Design and operating factors found to affect powerplant NO_x emissions include: fuel type and composition, heat release rate, burner configuration, excess air, and air inlet temperature. All of these factors, in turn, affect the previously mentioned basic factors (flame temperature, excess air, and residence time) which dictate the amount of NO_x formed. Low-excess air combustion, multi-burner combustors, steam and water injection, reduction of air preheating, and recirculation of stack gases methods have been used with some success on some steam boilers to reduce NO_x emissions. However, the use of these methods usually results in some degradation in operating efficiencies. Stack gas recirculation was used to reduce the temperature in the primary combustion zone and thereby achieve the aforementioned 150-ppm NO_x concentration in the exhaust of gas-fired steam boilers (Ref.223). Some, but obviously not all, of the above methods could be applied to reduce the NO_x emissions from gas turbine-type power systems without compromising good operating efficiencies.

In present-day gas turbines, the primary zone in the combustor is operated at near-stoichiometric fuel-air ratios, and a great deal of recirculation in the flame zone (not to be confused with stack gas recirculation used on steam boilers) is designed into the combustor to ensure proper combustion. These characteristics make for a high primary zone temperature and long residence time, both of which tend to promote the formation of NO_x. Operation of this type is necessary in aircraft jet engines to vaporize the fuel droplets and ensure stable flame propagation. It is generally difficult to have a lean fuel-air ratio in the primary zone, and to reduce recirculation in the aircraft application, and still meet the requirements for altitude restart capability and flame stability and propagation at low pressures (high altitude and idle power settings). In a stationary gas turbine power system, these requirements do not apply and stable flame propagation with a lean primary combustion zone and little recirculation could be achieved by premixing the fuel and air. This approach would not be considered for the aircraft application because the widely varying operating condition would enhance the possibility of an explosion. Another possible method of reducing NO_x emission in a stationary gas turbine-type power system involves water injection, which is used apparently to cool the primary combustion gases before substantial amounts of NO_x have time to form. It is claimed (Ref.224) that injecting water in a mass equal to the fuel mass would reduce NO_x emissions from gas turbines by 90%.

Although increases in turbine inlet (combustor exit) temperature would normally be expected to result in increased NO_x emissions, it has been shown that this is not necessarily so. It has been noted that the P&WA JT9D turbofan engine, which operates at higher turbine inlet temperatures than other commercial P&W turbofan engines, has generally lower NO_x emission concentrations. Apparently the annular construction of the JT9D combustor has a tendency toward lower NO_x emissions than do the annular combustor configurations used in other turbofan engines. Note that annular combustion chambers have been proposed for the second- and third-generation gas turbine-type power systems considered in this study. Although the gas turbine combustor inlet temperatures for the advanced engines considered in this study are generally higher than for present-day engines due to the use of higher compressor pressure ratios, the diluting effect of the inerts in the low-Btu fuel may lead to a lower primary zone temperature and lower NO_x emissions.

Based on the previous discussion, and the Corporate-sponsored work being conducted by UA Research Laboratories and Pratt & Whitney Aircraft, it appears that the principles and techniques required to reduce NO_x emissions from gas turbines are relatively well understood. Furthermore, it appears that the NO_x emissions achievable in advanced gas turbine-type power systems will be lower than in future steam boilers and that these lower NO_x emissions can be obtained without compromising gas turbine performance.

TABLE 107

REPRESENTATIVE UNIT TRAIN OPERATIONS FOR COAL

Millions of Tons/Year	Origin	Destination	Distance (Miles)	Rate ($)		Car Owner	Railroads	User
				Per Ton	Per Ton-Mile			
2	Southern Ohio	Detroit Michigan	311	2.01	0.0065	Coal Co.	PRR, NYC	Detroit Edison
1.5	Tunneltown, Pa.	Brunneris, Pa.	63		0.0045		PRR	Penna. Power & Light
	Tunneltown, Pa.	Martins Cr., Pa.	395	2.76	0.007		PRR	Penna Power & Light
	Egypt Valley, Ohio	West Olive, Mich.	468				B&O, C&O	Consumers Power
	Arco TN	Milledgeville, Ga.	457				SOU, CGa	Georgia Power
	Loverbridge, W. Va.	Bow, N. H.	868				P&LE, NYC, B&M	Public Service of NH
	Egypt Valley, Ohio	Lorain, Ohio	327				B&O	
1.5	Lynnville, Ind.	Hammond, Ind.	250	1.45	0.0058	Coal Co.	GM&O, NYC	Commonwealth Edison
4	Percy, Ill.	Joliet, Ill.	305	1.30	0.0023	Utility	GM&O	Commonwealth Edison
	Corbin, Ky.	Gary, Ind.	475				L&N, PRR	Commonwealth Edison
1.5		Wood River, Ill.	85			Utility		Illinois Power
1		Bailly, Ind.	360			Utility		No. Ind. Public Service
2		Mitchell, Ind.	325			R.R.		No. Ind. Public Service
1.5	Southern Ohio	Trenton Channel	275	1.57	0.0057	Utility		Detroit Edison
	No. Benton, Ill.	Machens, Mo.	133	1.20	0.009	R.R.	CB&Q	Union Electric
3.5	Lynch, Ky.	Corbin, Ky.	95				L&N	U.S. Steel

TABLE 108

PROPERTIES OF SOUTHERN ILLINOIS COAL, SEAM NO. 6

Based on Coal from Crown Mine, Farmersville, Illinois

Typical Analysis, wt %	
Moisture, as-received	13.0 (range 9 to 14)
Proximate Analysis, dry basis	
Volatile Matter	35.7
Fixed Carbon	54.3
Ash	10.0
Ultimate Analysis, dry basis	
Carbon	69.7
Hydrogen	5.3
Nitrogen	1.3
Sulfur	3.8 (range 3.3 to 4.3)
Oxygen	9.9
Ash	10.0
Ash Analysis	
SiO_2	47.0
Al_2O_3	20.0
Fe_2O_3	18.0
TiO_2	0.7
CaO	4.0
MgO	1.0
Na_2O	1.9
K_2O	2.4
SO_3	4.4
Chloride, wt % of dry coal	0.02 - 0.2
Ash Fusion Temperature, F	
Initial Deformation Temperature	1960
Softening Temperature	2020
Higher Heating Value, Btu/lb (dry basis)	12,700
Bulk Density, lb/ft^3 (dry basis)	43.8

TABLE 109

PROPERTIES OF VENEZUELAN RESIDUAL FUEL OIL

Based on Bechtel Report: Desulfurization Costs -
Residual Fuel Oil, February 1967

Ash, wt %	0.2
Sulfur, wt %	2.6
Metals[(1)], ppm	
Vanadium	425.3
Nickel	47.2
Iron	19.0
Sodium	8.0
Copper	0.3
Chromium	0.2
Viscosity, SUS @ 122 F	250
Viscosity, SUS @ 100 F	3700
Net Heating Value, Btu/gal	142,000
Gross Heating Value, Btu/gal	150,000
Flash, F (Pensky-Martens Closed Cup)	175

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(1) Metals content based upon crude and adjusted to residual specifications

TABLE 110

FIRST-GENERATION LURGI HOT-CARBONATE PROCESS

Equipment List

No. Reg'd.	Description	Equipment Purchased Cost-Dollars
5	Belt Conveyor, 76 TPH, 220 ft long CS-24 in.	90,000
2	Belt Conveyor, 200 TPH, 75 ft long, CS-24 in.	17,000
2	Underfeed Conveyors, 200 TPH, w/SS Hopper and Vibratory Feeders and Elevators	100,000
1	Raw Coal Conveyor Belt, 36 in. x 2000 ft long - 400 ft/min, 400 TPH	200,000
2	Primary Coal Crushers, 200 TPH, 85 hp	20,000
6	Briquetting Machines, 70 TPH	810,000
6	Distributing Conveyor - Piggy Back Belt Type, 18 in. x 50 ft, 70 TPH	30,000
6	Surge Hopper 100 Tons, 20 ft x 20 ft x 10 ft	70,000
23	Lurgi Dry Ash Gasifiers, 12 ft D, w/associated coal feeders, ash removal, water scrubbing and tar recycle	20,000,000*
2	Centrifugal Compressors, 3-stage, inter-cooled, 50,000 hp, 360,000 scfm at 320 psia	2,500,000
23	Waste-heat Boiler, 16.4 Million Btu/hr, 5200 ft^2	610,000
6	Gas Coolers, Duty 47.8 Million Btu/hr, 100 ft^2	70,000
6	Hot-Carbonate Absorption Towers 12 ft D x 20 ft H, CS Packed	210,000
6	Hot-Carbonate Stripping Towers, 12 ft D x 30 ft H, CS Packed	310,000
6	Liquor Circulation Pumps, 2200 gpm, 320 psi	90,000
6	Condenser, air cooler, forced draft, fined tube 12,000 ft^2	40,000
6	Reboilers - 20 Million Btu/hr, 2000 ft^2, CS/CS	50,000
1	Claus Conversion Plant, 205 LTPD plus storage	1,900,000*
1	Cooling Tower System, 2.4 million gph, forced draft, 25 F range, 10 F approach	600,000*
1	Water treatment plant, 100 million gph	400,000*
1	Gas-fired package boiler, 300,000 lb/hr steam	2,200,000*

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* Total installed cost including commodities, indirect costs, and contingency.

TABLE 111

SECOND-GENERATION PARTIAL-OXIDATION HOT-CARBONATE PROCESS

Equipment List

No. Req'd.	Description	Equipment Purchased Cost-Dollars
2	Underfeed Conveyors 115 TPH, w SS Hopper and Vibrating Feeders	35,000
1	Raw Coal Conveyor-Belt, 30 in. x 2000 ft long - 400 ft/min, 310 TPH	175,000
4	Belt Conveyor, 80 TPH, 220 ft long CS-24 in.	20,000
4	Primary Coal Crusher, 180 TPH, 70 hp	20,000
4	Hammer Mill, 80 TPH, 150 hp	105,000
4	Crushed Coal Elevators, 80 TPH	130,000
4	Surge Hopper, 100 T, 20 ft x 20 ft x 10 ft	50,000
4	Distributing Conveyor - Piggy Back Belt Type, 18 in. x 50 ft, 70 TPH	30,000
4	Slurry Tank, 14,000 gal, SS clad, 2/agitation	98,000
4	Slurry Pump, centrifugal, 450 gph, 85 hp	35,000
4	Vaporizer, 117 million Btu/hr, SS/CS, 7600 ft^2	171,000
4	Coal-Steam Separator, 47,000 acfm	14,000
4	Texaco Partial Oxidation Gasifiers, 14.75 ft D x 35.5 ft H, 18 in. refractory lining	700,000
4	Steam Coal Preheater, 64 million Btu/hr, SS/SS 21,000 ft^2	310,000
4	Pebble Elevator, 140 TPH, 80 ft H	51,000
4	Air Preheater, 17.75 ft D x 55 ft H, 13 in. refractor lining, 140 million Btu/hr pebble bed	1,300,000
4	Waste Heat Boiler, CS/CS, 60 million Btu/hr, 7300 ft^2	165,000
2	Air Compressor, 3-stage, intercooled, 280,000 scfm @ 256 psia, 34,000 hp	1,800,000
4	Multiclone Banks, 20,000 acfm	130,000
4	Venturi Scrubbers, 14,000 acfm	36,000
4	Water Scrubbing Towers, 12.3 ft D x 24 ft H, 1450 ft^3 drip-grind packing	228,000
4	Ash Separator Tanks, CS, 13,000 gal	11,000
8	Water Circulation Pumps	40,000
4	Hot-Carbonate Absorption Towers, 67 ft D x 27 ft H, CS, 15 turbogrid trays	90,000
4	Hot-Carbonate Stripping Towers, 6.7 ft D x 52 ft H, CS 28 turbogrid trays	160,000
4	Condensers, air cooled, 22,000 ft^2 of fin tube	58,000
4	Separators 6 ft D x 20 ft L, CS Horizontal	25,000
4	Demisters 6 ft D, CS	10,000
4	Liquor Pumps, 2000 gph, 200 psi head, SS internals	200,000
1	Waste Water Stripping Tower, 10.5 ft D x 30 ft H, 20 trays	54,000
1	Air Blower for Waste Water Stripping, 200,000 cfm @ 5 ft H_2O	75,000
1	Cooling Tower System, forced draft, 500,000 gph, 25 F range, 10 F approach	150,000*
1	Process Water Treatment Plant, 60,000 gph	200,000*

---

* Total installed cost including commodities, indirect costs, and contingency.

TABLE 112

SECOND-GENERATION IRON OXIDE SULFUR REMOVAL PROCESS

Equipment List

No. Req'd	Description	Equipment Purchased Cost-Dollars
4	Absorber, 12.5 ft D x 165 ft H, 5-stage fluidized bed, 11-in. refractory lining	1,550,000
4	Regenerator, 7.5 ft D x 140 ft H, 4-stage fluidized bed, 11-in. refractory lining	600,000
4	Regeneration Air Compressor, 3-stage, inter-cooled 34,000 hp	96,000
1	Regenerator Off-Gas Filter, 11 ft D x 12 ft H, brick line	40,000
1	Dust Removal System, rotating stream separators plus high-velocity filtration	143,000

TABLE 113

THIRD-GENERATION PARTIAL-OXIDATION HOT-CARBONATE PROCESS

Equipment List

No. Reg'd	Description	Equipment Purchased Cost-Dollars
2	Underfeed Conveyors 145 TPH w SS Hopper and Vibrating Feeders	34,000
1	Raw Coal Conveyor-Belt, 30 in. x 2000 ft long - 400 ft/min, 290 TPH	168,000
2	Belt Conveyor, 145 TPH, 220 ft long CS - 24 in.	19,000
2	Primary Coal Crusher, 145 TPH, 120 hp	101,000
2	Crushed Coal Elevator, 145 TPH	125,000
2	Surge Hopper, 170 T, 25 ft x 25 ft x 10 ft	48,000
2	Distributing Conveyor - Piggy Back Belt Type 24 in. x 20 ft, 145 TPH	29,000
2	Slurry Tank, 26,000 gal, SS clad, w/agitation	94,000
2	Slurry Pump, centrifugal, 850 gph, 270 hp	70,000
2	Vaporizer, 220 million Btu/hr, SS/CS, 13,500 ft^2	158,000
2	Coal-Steam Separator, 48,000 acfm	10,000
2	Texaco Partial Oxidation Gasifiers, 12.75 ft x 29 ft H, 18 in. refractory lining	500,000
2	Steam Coal Preheater, 120 million Btu/hr, SS/SS 37,500 ft^2	390,000
2	Pebble Elevator, 265 TPH, 80 ft H	45,000
2	Air Preheater, 23.0 ft x 55 ft H, 13 in. refractor lining, 265 million Btu/hr, pebble bed	1,245,000
2	Waste Heat Boiler, CS/CS, 113 million Btu/hr, 13,000 ft^2	160,000
2	Air Compressor, 3 stage, intercooled, 258,000 scfm @ 355 psia, 36,000 hp	1,910,000
2	Multiclone Banks, 22,000 acfm	70,000
2	Venturi Scrubbers, 16,000 acfm	20,000
2	Water Scrubbing Towers, 13 ft D x 40 ft H, 2500 ft^3 drip-grid packing	205,000
2	Ash Separator Tanks, CS, 20,000 gal	16,000
4	Water Circulation Pumps	30,000
2	Hot-Carbonate Absorption Tower, 7.5 ft D x 60 ft H, CS, 30 turbogrid trays	123,000
2	Hot-Carbonate Stripping Tower, 7.5 ft D x 100 ft H, 60 turbogrid trays	210,000
2	Condenser, air cooled, 44,000 ft^2 of fin tube	58,000
2	Separators 8.5 ft x 20 ft L, CS Horizontal	25,000
2	Demisters 8.5 ft, CS	10,000
2	Liquor Pumps, 4000 gpm, 325 psi head, SS internals	270,000
1	Waste Water Stripping Tower, 10.5 ft D x 30 ft H, 20 trays	54,000
1	Air Blower for Waste Water Stripping, 200,000 cfm @ 5 in. H_2O	75,000
1	Cooling Tower System, forced draft, 1.3 million gph, 25 F range, 10 F approach	310,000*
1	Process Water Treatment Plant, 100,000 gph	440,000*
1	Recycle Gas Compressor, centrifugal, 3 stage intercooled, 36,000 scfm @ 326 psi, 14,000 hp	61,000
1	Package Boiler, gas fired, 40,000 lb/hr	320,000*

---

* Total installed cost including commodities, indirect costs, and contingency.

TABLE 114

THIRD-GENERATION DOLOMITE ACCEPTOR SULFUR REMOVAL SYSTEM

Equipment List

No. Req'd	Description	Equipment Purchaged Cost-Dollars
2	Desulfurizer, fluidized bed, 20 ft D x 20 ft H, 8 in. refractory lining, internal cyclone	560,000
1	Regenerator, fluidized bed, 7 ft D x 20 ft H, 6 in. refractory lining, internal cyclone	50,000
1	Calciner, fluidized bed, 14 ft D x 40 ft H, 18 in. refractory lining, internal cyclone	240,000
1	Catacarb Scrubbing System, 20,000 actual ft^3/min	1,500,000*
1	Recycle Gas Compressor, centrifugal, 1620 hp	660,000

---

* Total installed cost including commodities, indirect costs, and contingency.

TABLE 115

RAW MATERIAL COSTS AND BY-PRODUCT CREDITS

Raw Material	Location US	Venezuela
Residual Oil, $/million Btu	30.0	-
Natural Gas, ¢/million Btu	55.0	6.0
Coal, ¢/million Btu	20.0	-
Dolomite, $/ton	5.0	-
By-Product		
Sulfur, $/long ton	25.0	25.0
Gasoline, ¢/gal	11.0	6.5
Middle Distillate, ¢/gal	9.0	8.0
Tar Acids, ¢/lb	4.0	-
Ammonia, $/ton	35.0	-
Sulfuric Acid, $/ton	10.0	-

TABLE 116

DUST COLLECTOR CHARACTERISTICS

Collector Type	Smallest (1) Particle Size Collected-Microns	Remarks
A. Settling Chamber	10 - 40	Multiple tray required for small size collection
B. Inertial Separators		
1. Baffle Chamber	20	
2. High Vel. Impaction	1	High efficiency, compact
3. Louver Type	10	Possible abrasion problem
4. Gas Reversal	40	Used as a precleaner
5. Rotating	5	Compact, temperature-limited
C. Cyclones	5 - 15	Small multicyclones requires for small particle sizes
D. Filters	0.1	High efficiencies but large surfaces are required
E. Electrostatic Precipitator	0.1	High-efficiency, expensive

---

(1) At 90 - 95% efficiency, by weight

TABLE 117

COMPOSITION OF REPRESENTATIVE LOW-BTU FUEL GASES

	Volume %				
Constituent	Fuel Gas I(1)	Fuel Gas II(2)	Fuel Gas III(3)	Fuel Gas IV(4)	Fuel Gas V(5)
H_2	14.50	24.05	25.50	21.70	24.30
CO	27.40	16.35	27.80	25.30	25.80
CO_2	9.80	0.77	1.00	4.90	0.90
CH_4	2.60	6.59	0.50	0.40	0.40
H_2O	0.20	5.90	7.50	7.00	0.60
N_2	48.3	46.18	37.70	40.70	48.00
H_2S	0.16	0.16	-	0.008	0.01
COS	-	-	-	0.023	-

---

(1) Preliminary estimate for Wellman-Galusha coal gasification. See Table 10 for revised estimate.

(2) Preliminary estimate for Lurgi coal gasification. See Table 6 for revised estimate.

(3) Preliminary estimate for Texaco coal gasification. See Table 18 for revised estimate.

(4) Partial oxidation of coal with high-temperature dolomite sulfur removal. See Table 32.

(5) Texaco gasification of residual oil. See Table 23.

TABLE 118

COMPARATIVE PERFORMANCE WITH REPRESENTATIVE FUEL GASES

Second-Generation COGAS Power Systems
Turbine Inlet Temperature = 2800 F
Compressor Pressure Ratio = 12
Cooling Airflow = 8.5%
Cooling Air Temperature = 200 F

	Methane	Fuel Gas I	Fuel Gas II	Fuel Gas III	Fuel Gas IV	Fuel Gas V
Fuel Supply Temperature, F	80	100	230	232	1500	200
Total Heat Content, Btu/ft^3	1013	163	202	181	194	169
Deviation from Mean Power System Efficiency, %	5.6	0.75	2.4	0.19	1.3	0.37

TABLE 120

ILLUSTRATIVE ECONOMIC RESULTS FOR A 150-MW, SECOND-GENERATION GAS TURBINE ENGINE

Manufacturing Costs

Item	Cost - $
Low-Pressure Compressor	
Disks	11,154
Blades	5,225
Vanes	9,057
Hubs and Spacers	5,535
High-Pressure Compressor	
Disks	29,647
Blades	6,642
Vanes	12,087
Hubs and Spacers	8,914
Burner	37,474
High-Pressure Turbine	
Disks	6,179
Blades	31,292
Vanes	22,150
Low-Pressure Turbine	
Disks	8,272
Blades	125,961
Vanes	38,833
Power Turbine	
Disks	191,252
Blades	138,507
Vanes	58,090
Shafts	14,610
Bearings	17,515
Casings	21,330
Miscellaneous Parts	175,551

TABLE 120 (CONT'D)

Item	Cost - $
Assembly and Test	90,000
X-ray Cost	1,500
Total Manufacturing Cost	1,066,780
Total Selling Price	2,251,800

TABLE 121

ILLUSTRATIVE GEOMETRIC RESULTS FOR A 150-MW, SECOND-GENERATION GAS TURBINE ENGINE

Low-Pressure Compressor

Stage No.	1	2	3	4	5	6
No. of Blades	39	42	45	50	57	69

High-Pressure Compressor

Stage No.	1	2	3	4	5	6	7	8	9	10
No. of Blades	46	48	49	51	54	57	62	68	77	92

Overall Engine

Section	Length (ft)
Low-Pressure Compressor	3.74
High-Pressure Compressor	4.19
Burner	1.43
High-Pressure Turbine	1.11
Low-Pressure Turbine	1.23
Transition Case	1.70
Power Turbine	6.12
Overall Engine	19.54

TABLE 119

COMPRESSOR AND TURBINE VANE, BLADE, AND DISK MATERIALS FOR ANALYTICAL MODEL OF ADVANCED GAS TURBINE ENGINES

Vane and Blade Materials

Name
Waspaloy
Udimet 500
Udimet 700
Incoloy 901 (AMS 5661)
Ti-8Al-1Mo-IV
Ti-6Al-2Sn-4Z4-2Mo
B-1900A
Ti-6Al-4V (AMS 4928)
Greek Ascoloy (AMS 5616)

Disk Materials

Incoloy 901 (AMS 5661)

Templex (AMS 6304)

RATES FOR COAL SHIPMENT BY RAIL

COAL ENERGY EQUIVALENT ASSUMED 25,000,000 BTU/TON

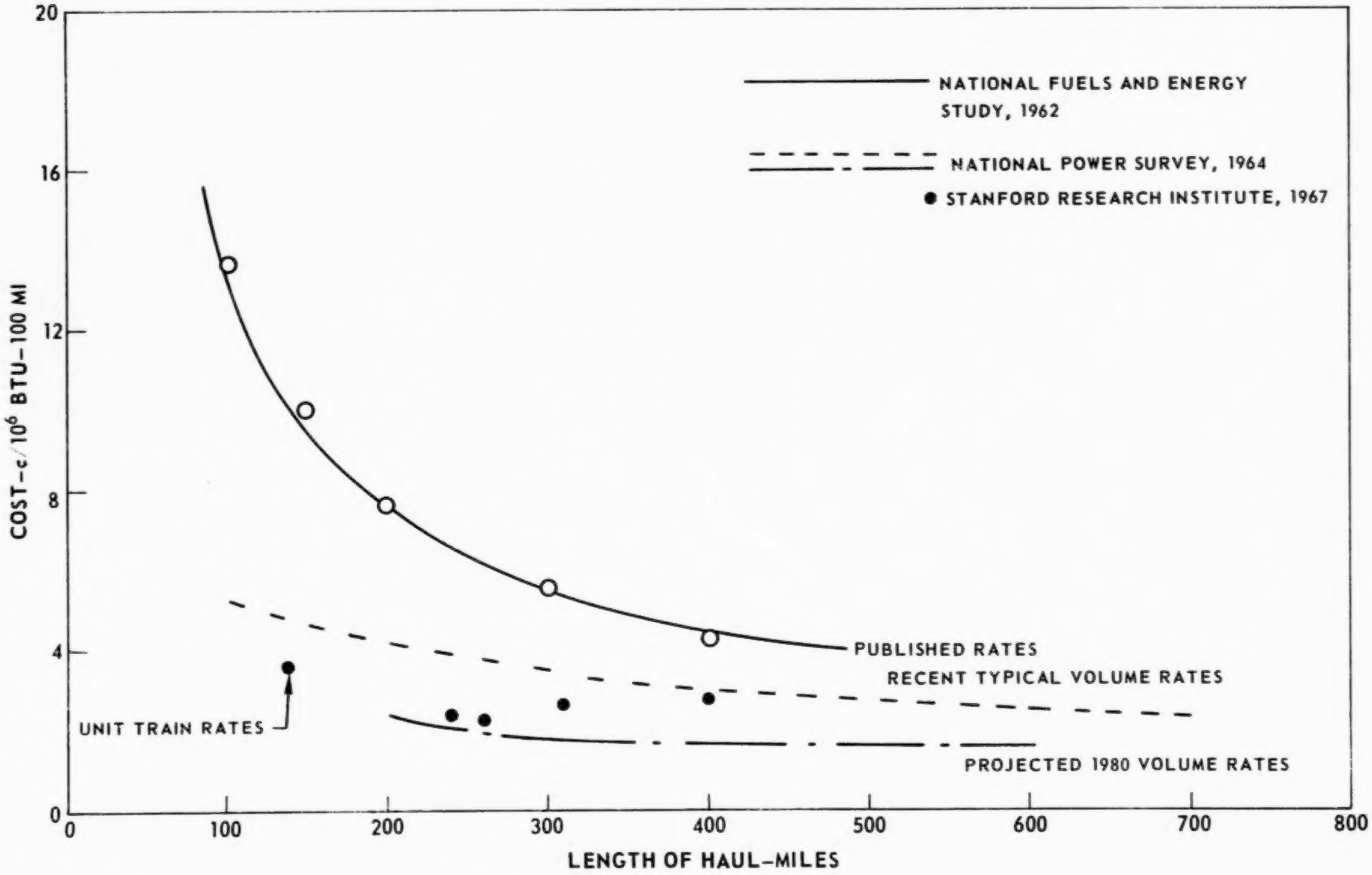


FIG. 150

FIG. 151

EFFECT OF TRANSPORTATION CHARGES ON DIFFERENTIAL BUSBAR POWER COST FOR 1000 MW STATION

1980 TECHNOLOGY

DIFFERENTIAL COST = (COST USING 1000 BTU/FT^3 GAS PIPELINED FROM COAL CONVERTER)–(COST USING LOW-HEAT-CONTENT GAS MADE FROM COAL RAIL HAULED TO POWERPLANT)

PIPELINE GAS

BASE COST = 40¢/MILLION BTU

TRANSPORTATION COST = 2¢/MILLION BTU–100 MI

LOW-HEAT-CONTENT GAS

38¢/MILLION BTU

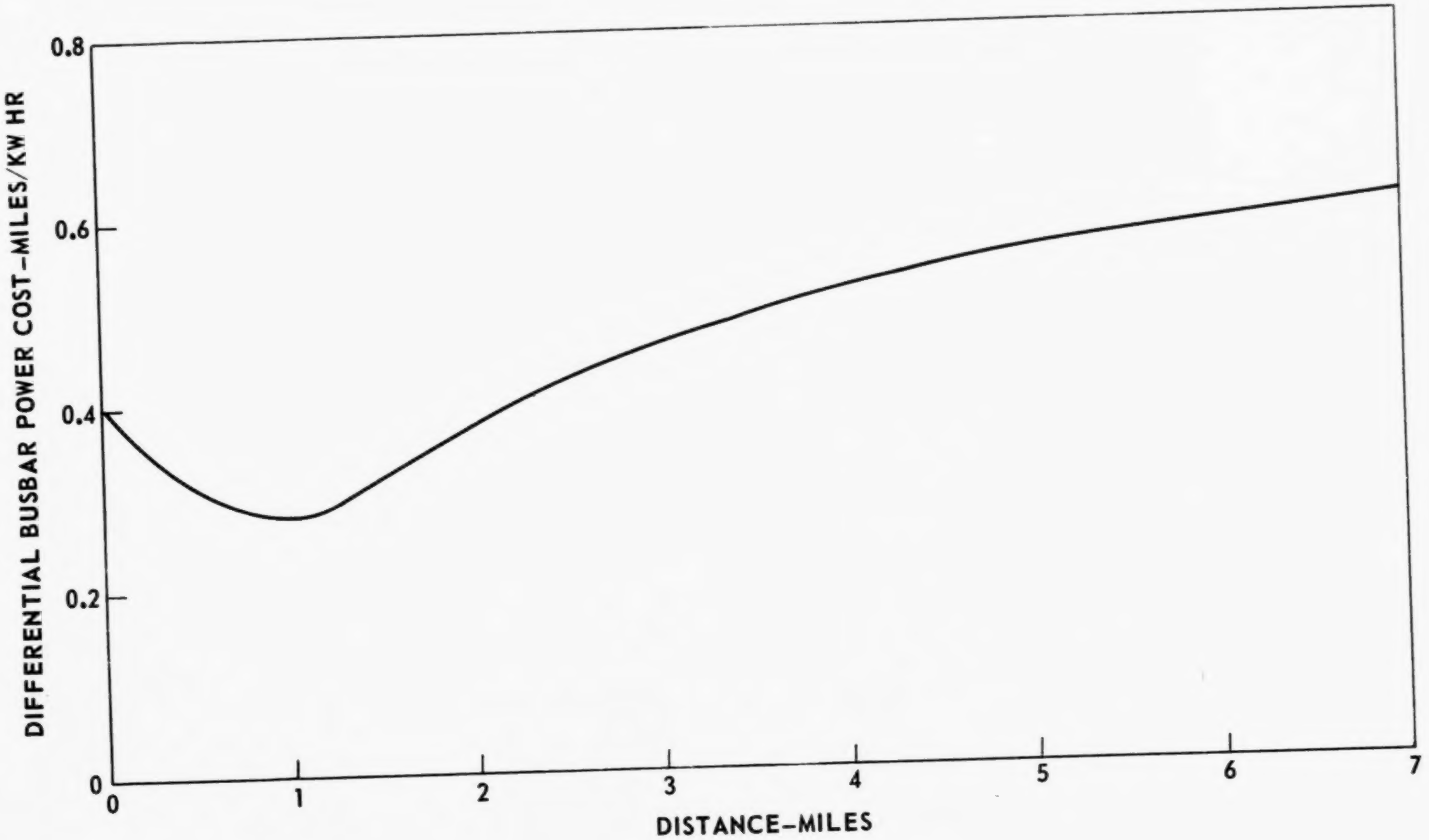


FIG. 152

FUEL OIL TRANSPORTATION COSTS

NATIONAL FUELS AND ENERGY STUDY, 1962

BASED ON RESIDUAL OIL ENERGY EQUIVALENT OF 150,000 BTU/GAL

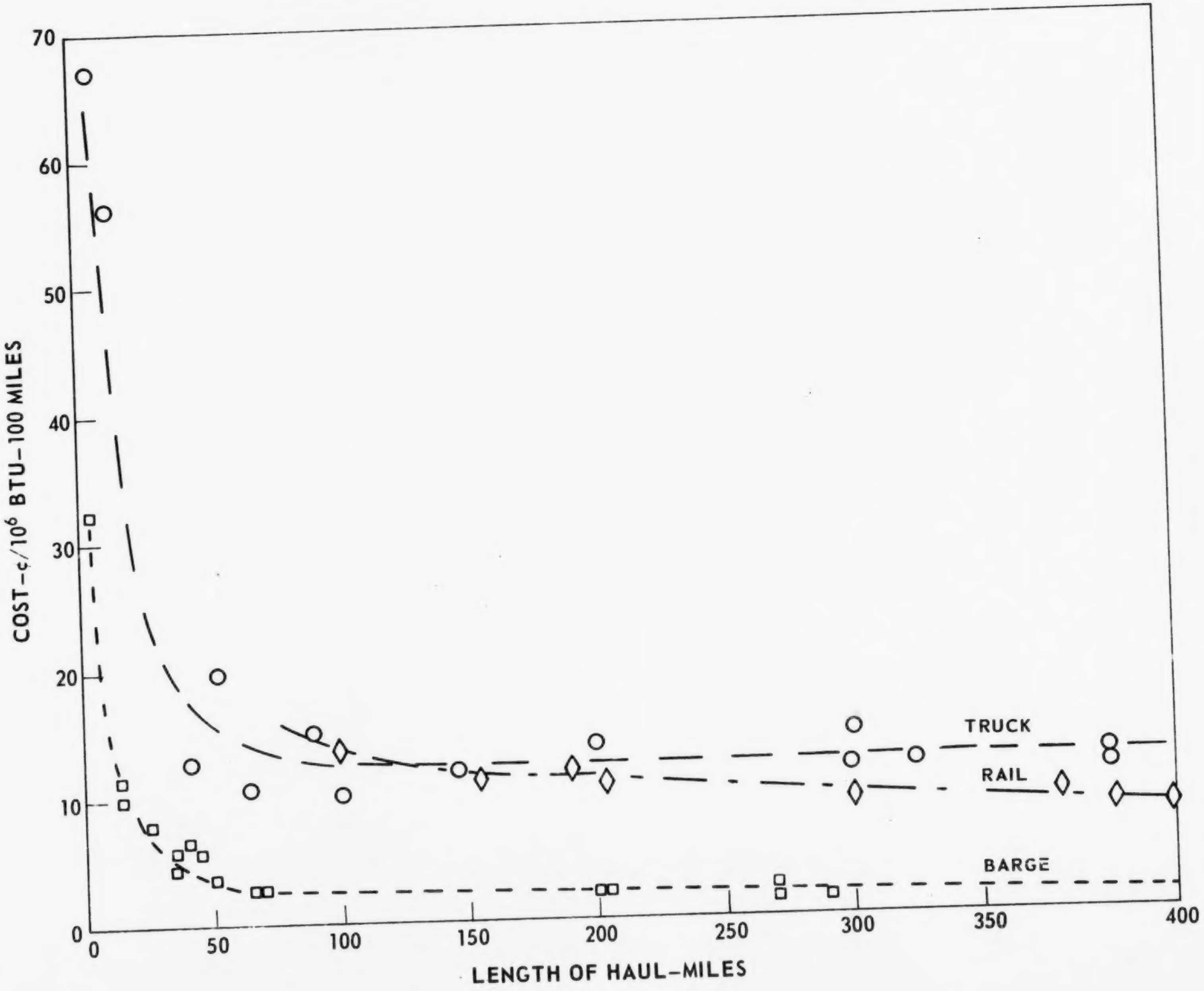
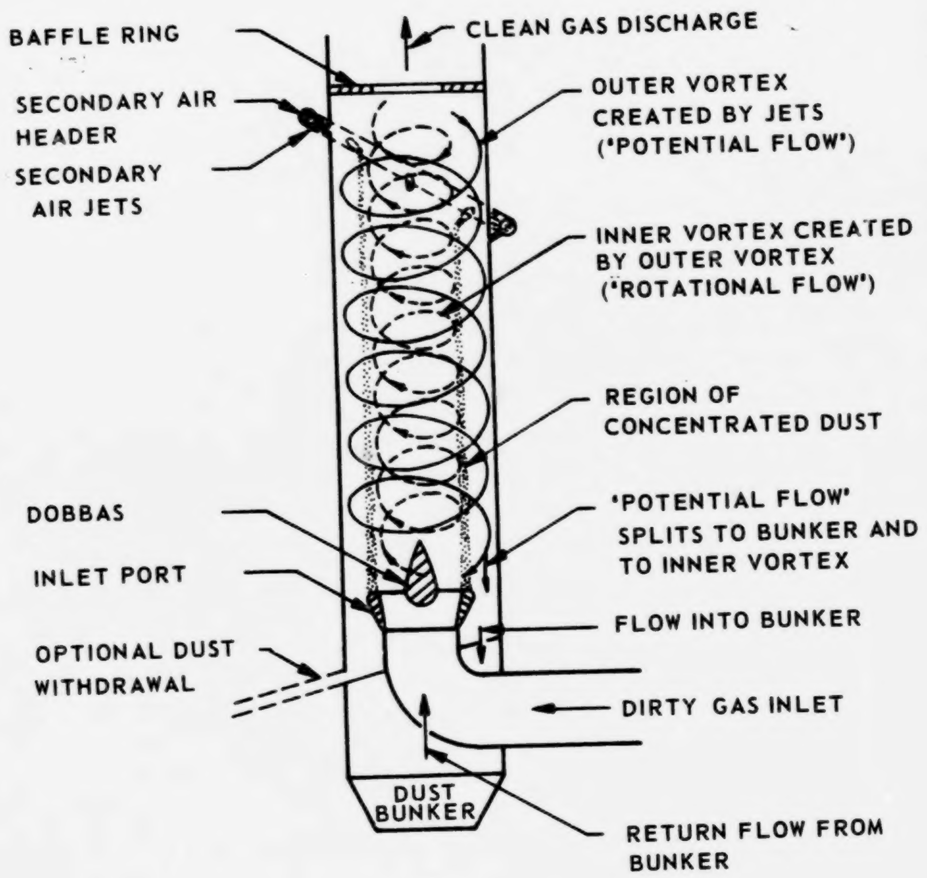


FIG. 153

TYPICAL ROTARY STREAM DUST SEPARATOR AND MAJOR GAS FLOWS


BAFFLE RING
CLEAN GAS DISCHARGE
SECONDARY AIR HEADER
OUTER VORTEX CREATED BY JETS ('POTENTIAL FLOW')
SECONDARY AIR JETS
INNER VORTEX CREATED BY OUTER VORTEX ('ROTATIONAL FLOW')
REGION OF CONCENTRATED DUST
DOBBAS
'POTENTIAL FLOW' SPLITS TO BUNKER AND TO INNER VORTEX
INLET PORT
FLOW INTO BUNKER
OPTIONAL DUST WITHDRAWAL
DIRTY GAS INLET
DUST BUNKER
RETURN FLOW FROM BUNKER

MAJOR DIMENSIONS OF EXPERIMENTAL 8-INCH-DIAMETER TUBE

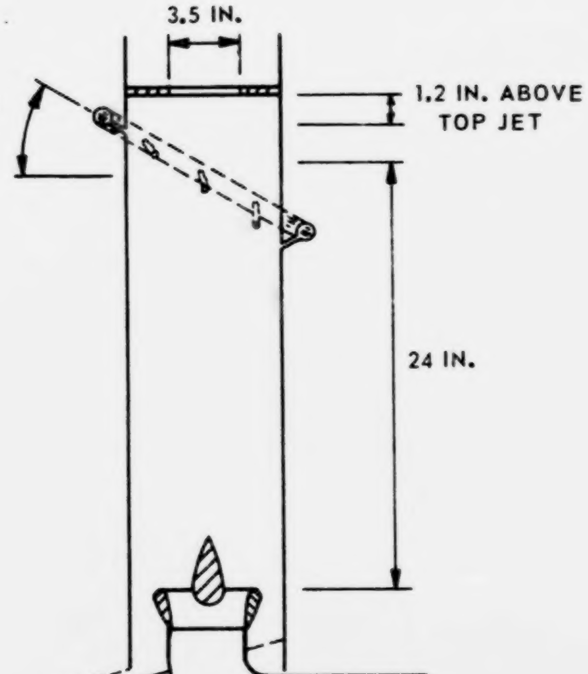
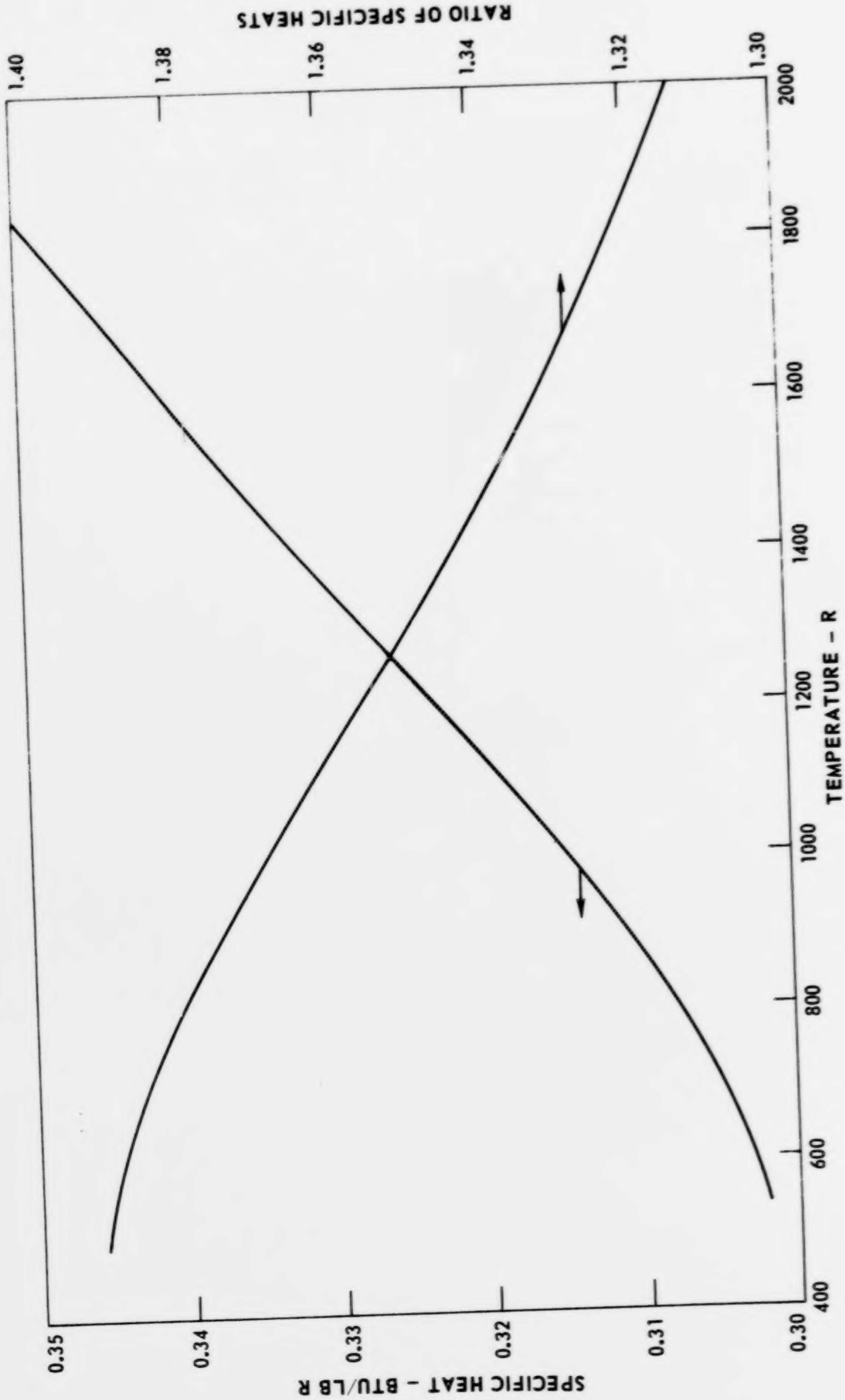


FIG. 155



SPECIFIC HEAT AND SPECIFIC HEAT RATIO OF FUEL GAS I

TEMPERATURE RISE FOR FUEL GAS I

FUEL TEMPERATURE = 80 F

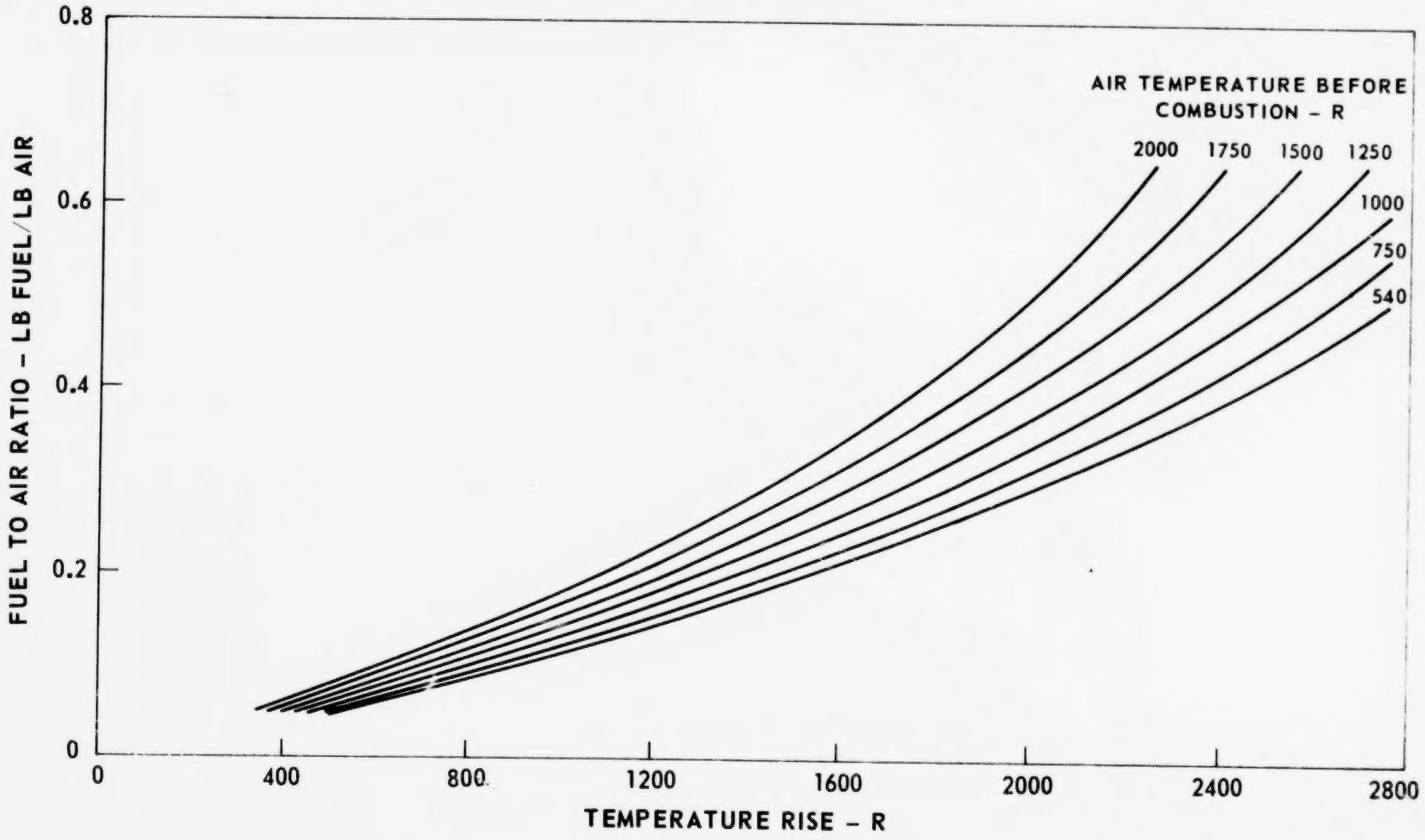
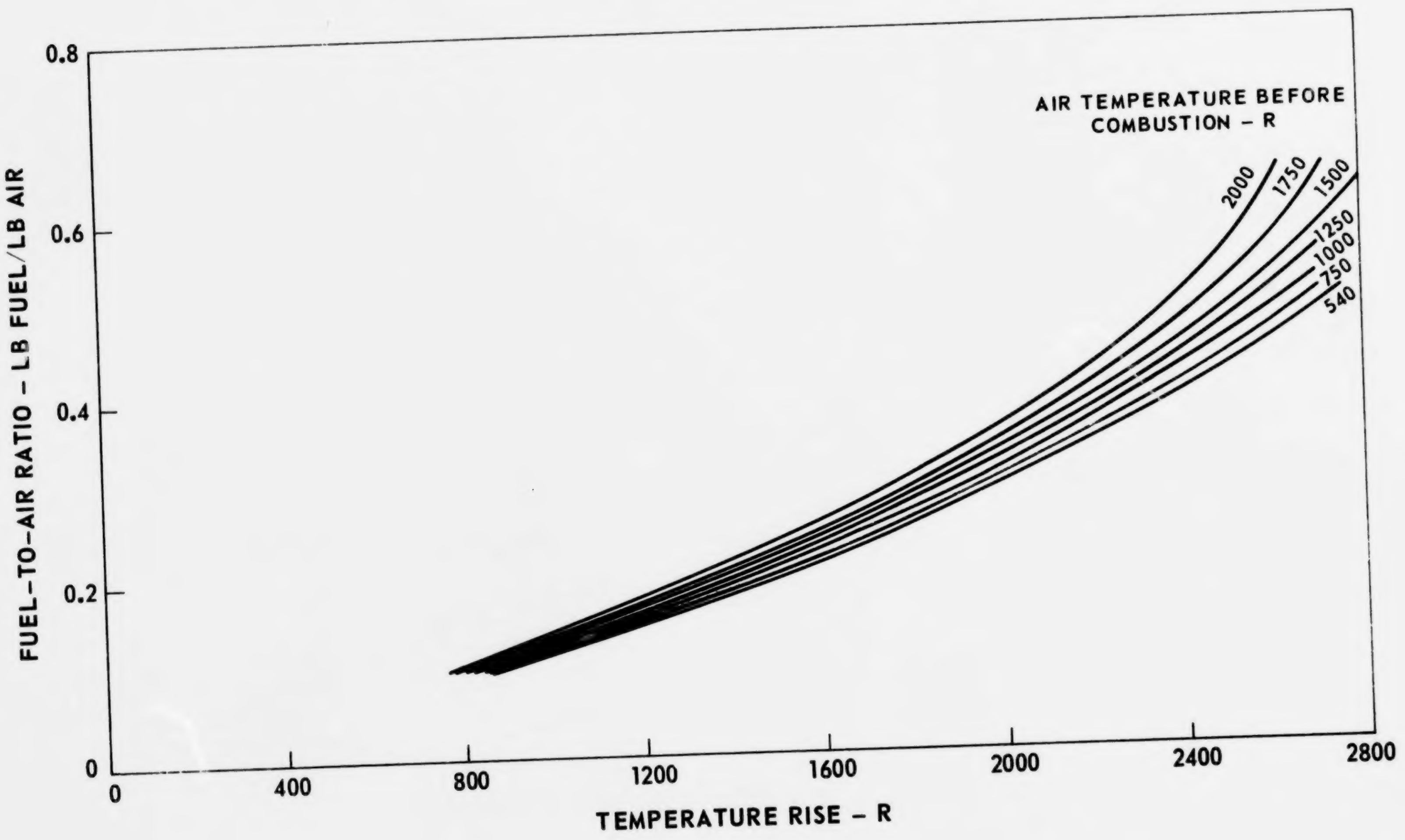


FIG. 156

FIG. 157

TEMPERATURE RISE OF FUEL GAS I

FUEL TEMPERATURE EQUAL TO AIR TEMPERATURE



SPECIFIC HEAT OF PRODUCTS OF COMBUSTION OF FUEL GAS I

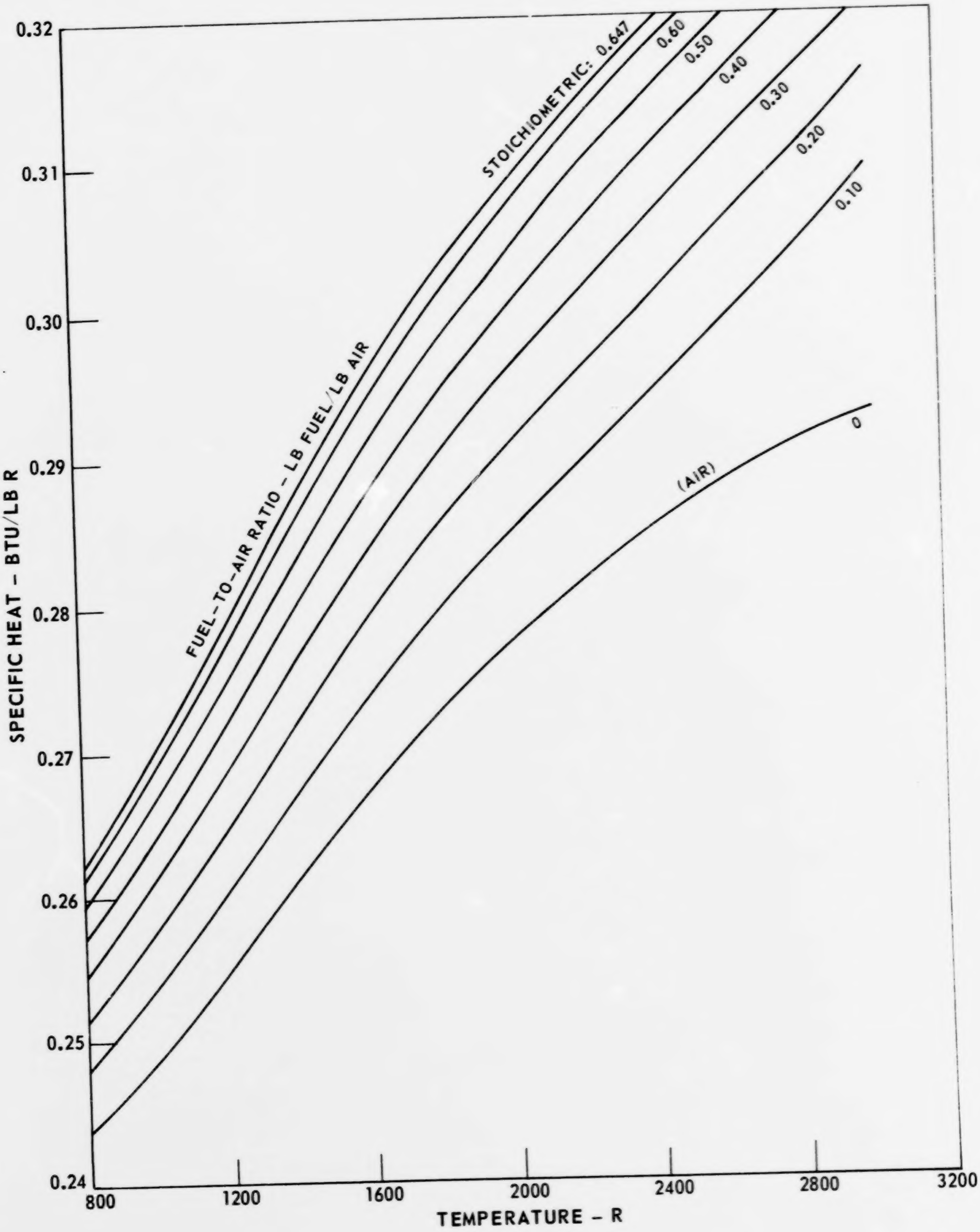


FIG. 159

MOLECULAR WEIGHT AND GAS CONSTANT OF PRODUCTS OF COMBUSTION OF FUEL GAS I

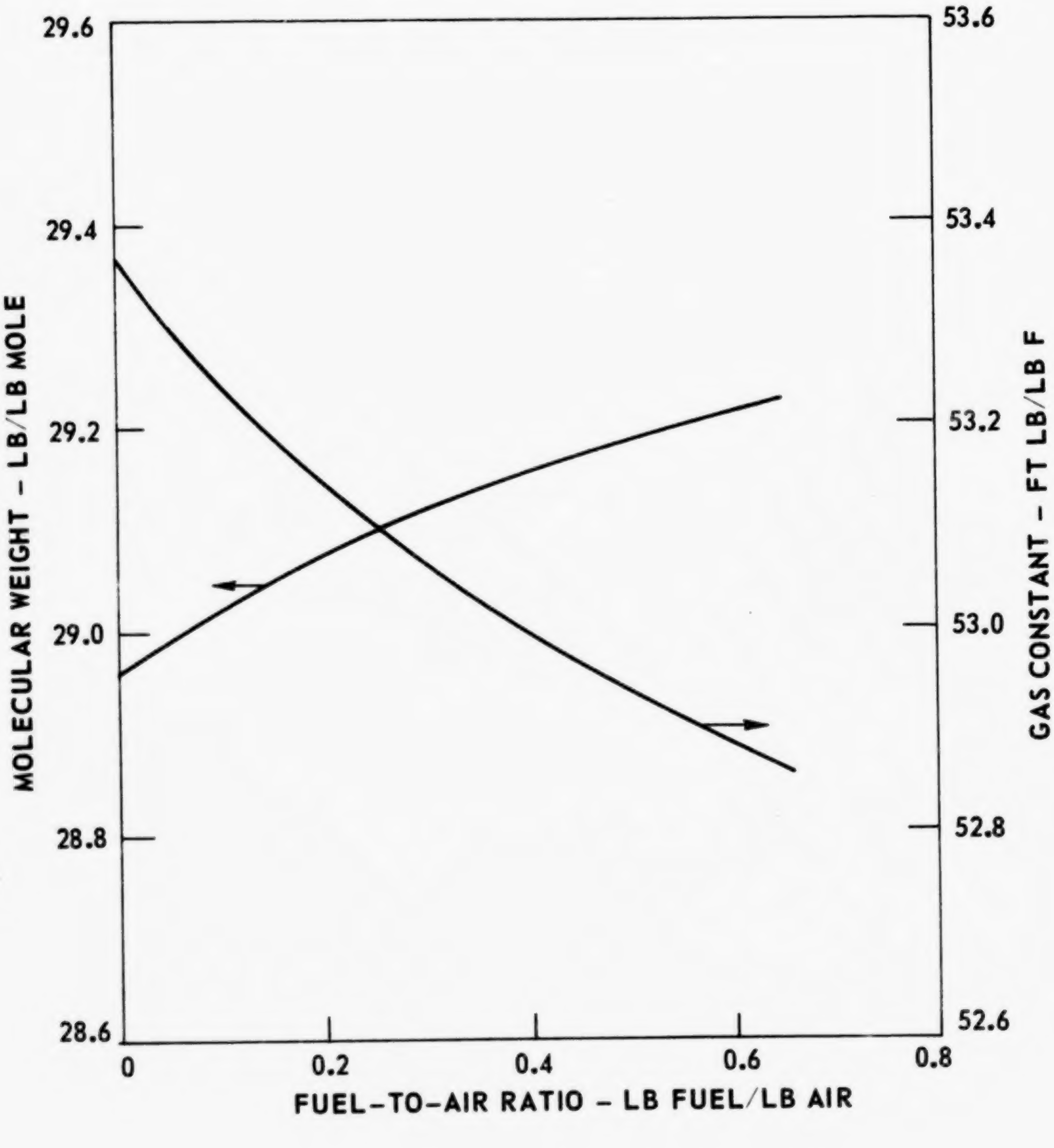


FIG. 160

SPECIFIC HEAT AND SPECIFIC HEAT RATIO OF FUEL GAS II

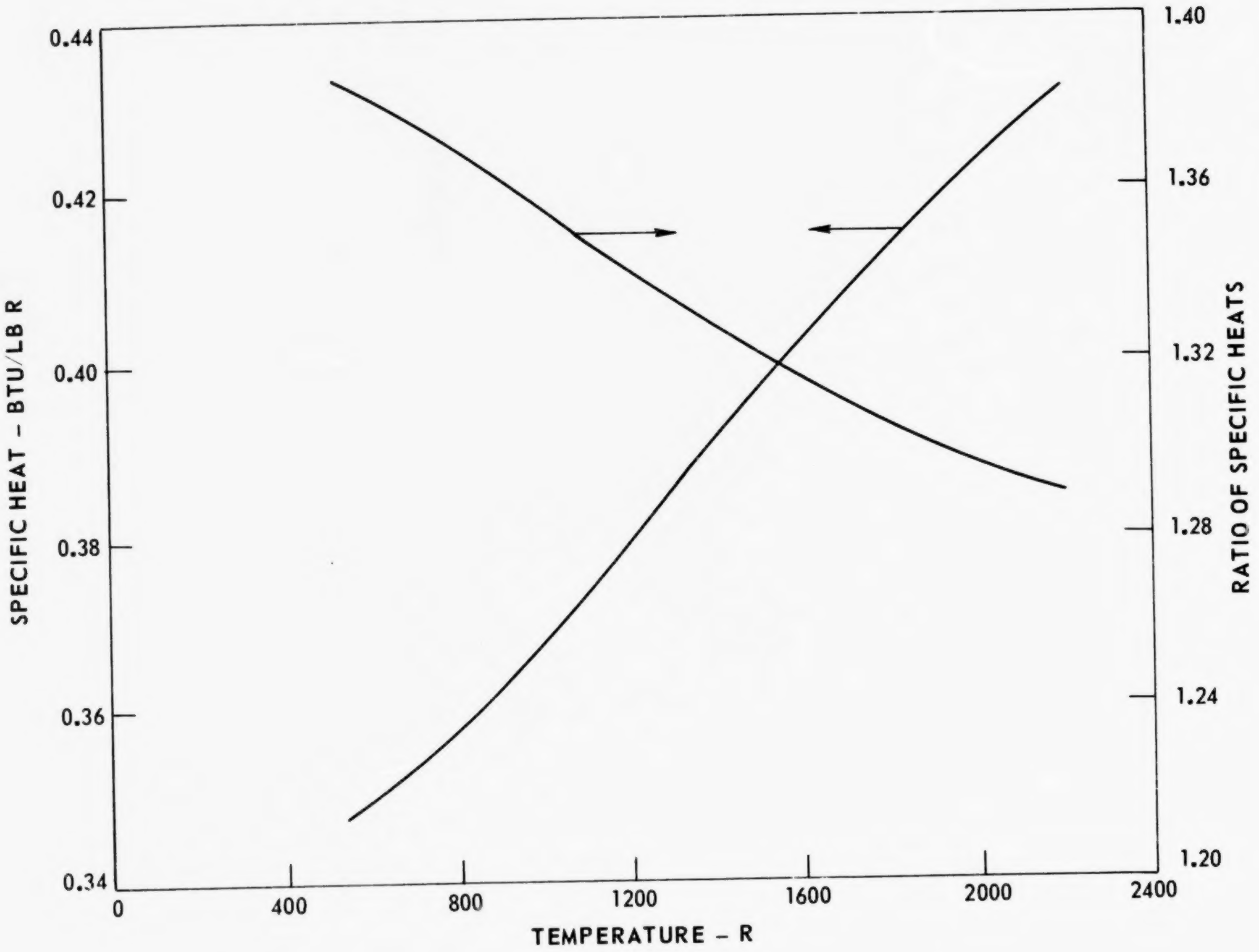
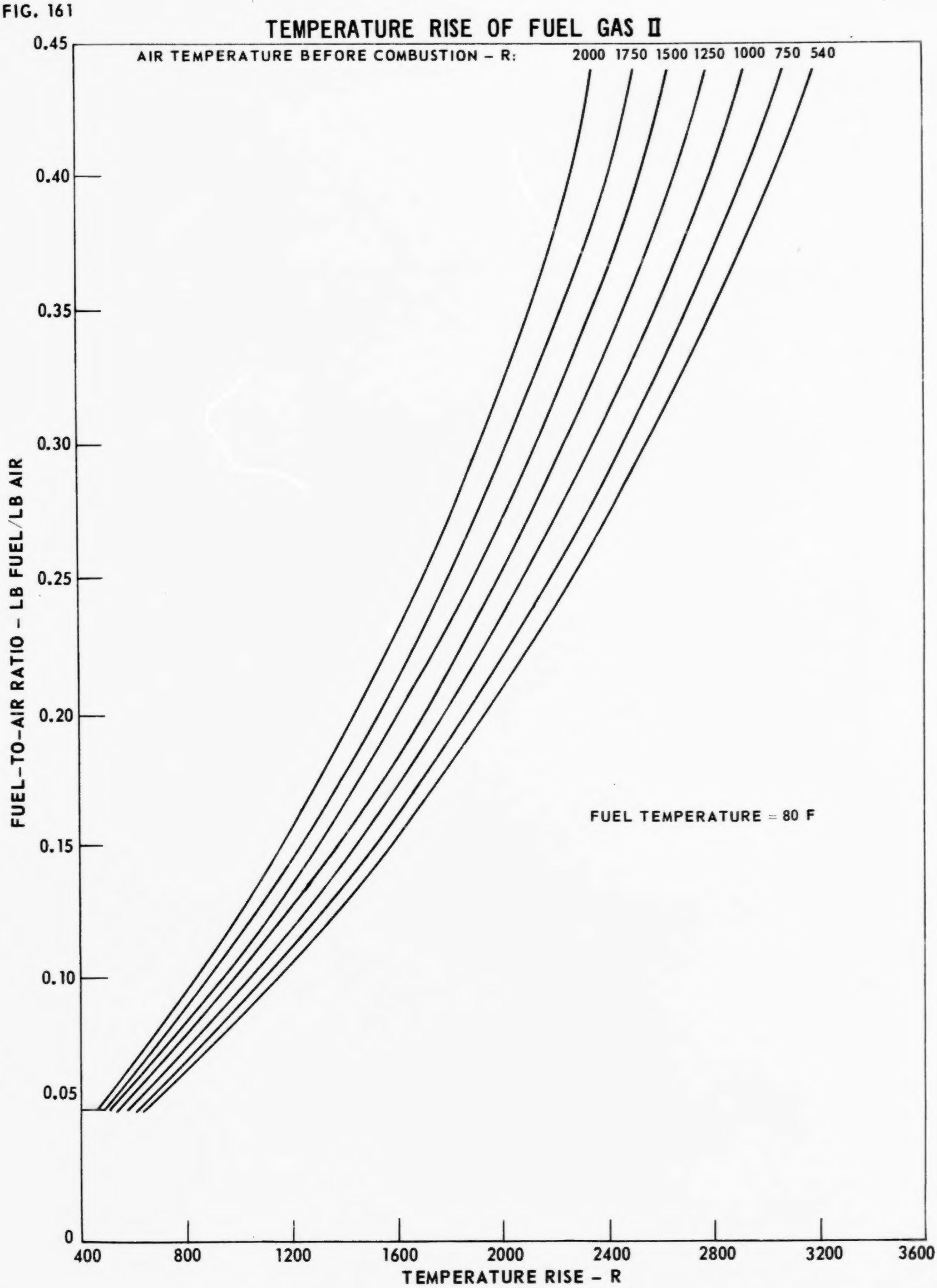


FIG. 161



TEMPERATURE RISE OF FUEL GAS II
AIR TEMPERATURE BEFORE COMBUSTION – R:
2000
1750
1500
1250
1000
750
540
FUEL TEMPERATURE = 80 F
FUEL-TO-AIR RATIO – LB FUEL/LB AIR
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
400
800
1200
1600
2000
2400
2800
3200
3600
TEMPERATURE RISE – R

FIG. 162

TEMPERATURE RISE OF FUEL GAS II

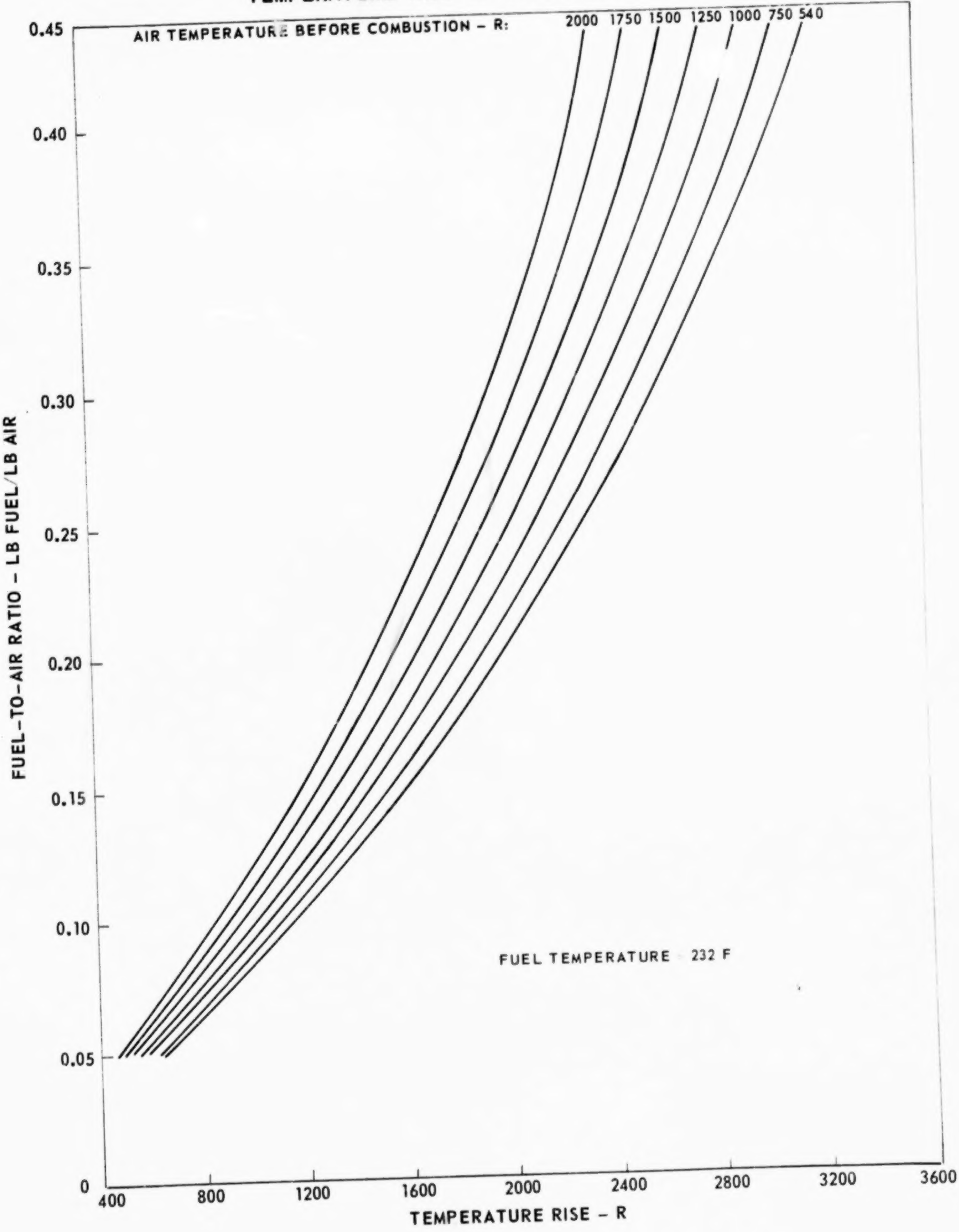


FIG. 163

TEMPERATURE RISE OF FUEL GAS II

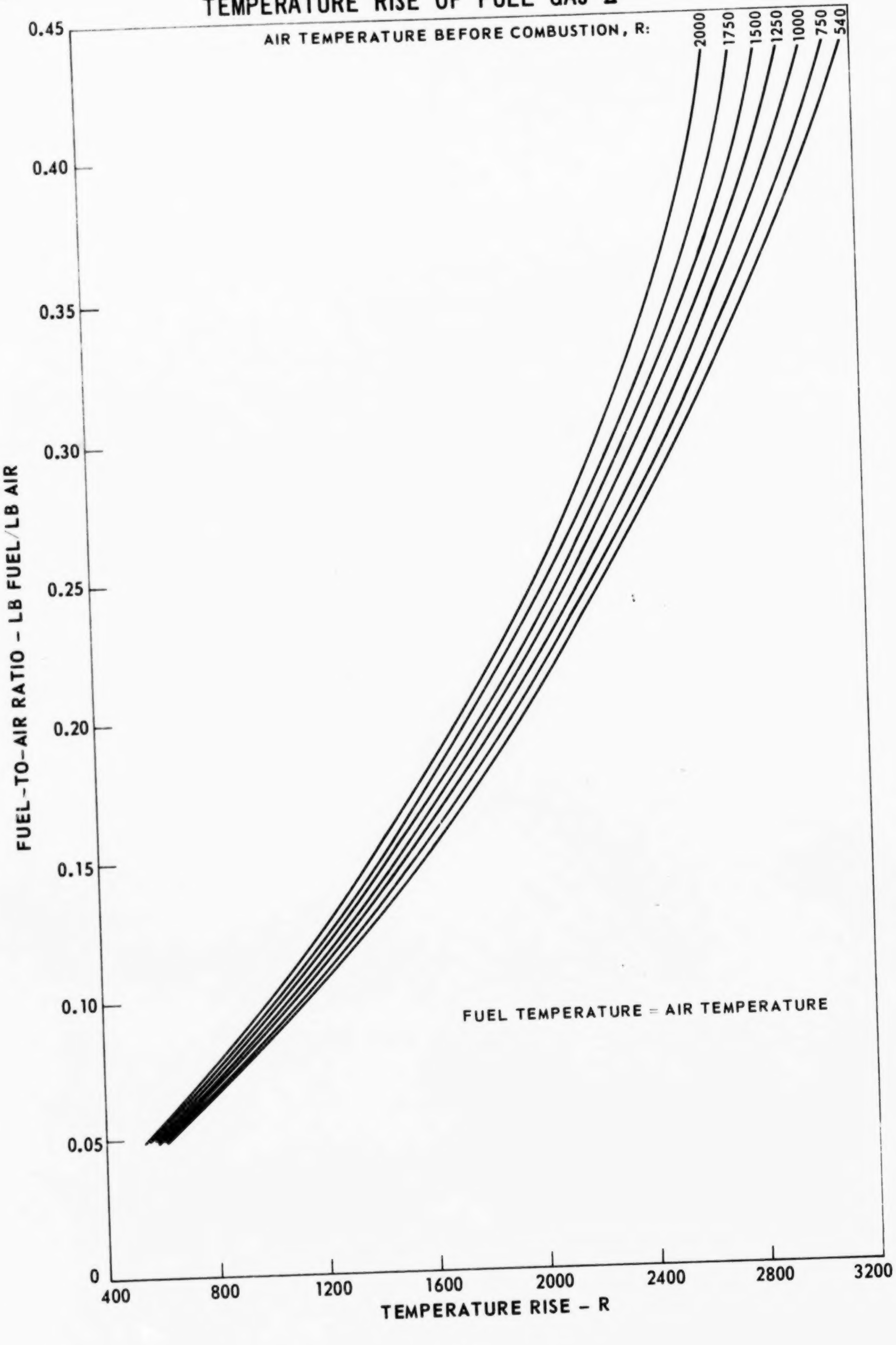


FIG. 164

SPECIFIC HEAT OF PRODUCTS OF COMBUSTION OF FUEL GAS II

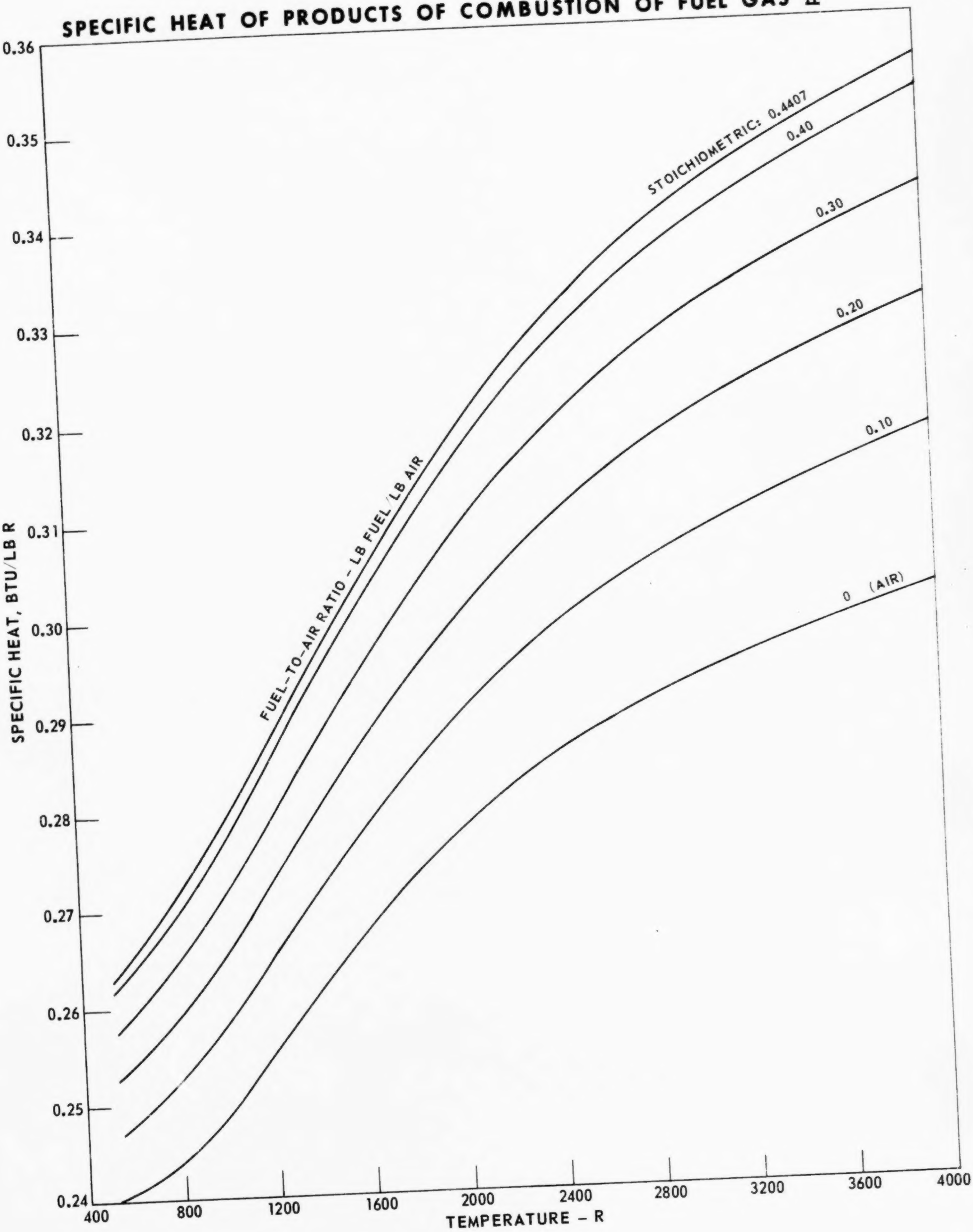
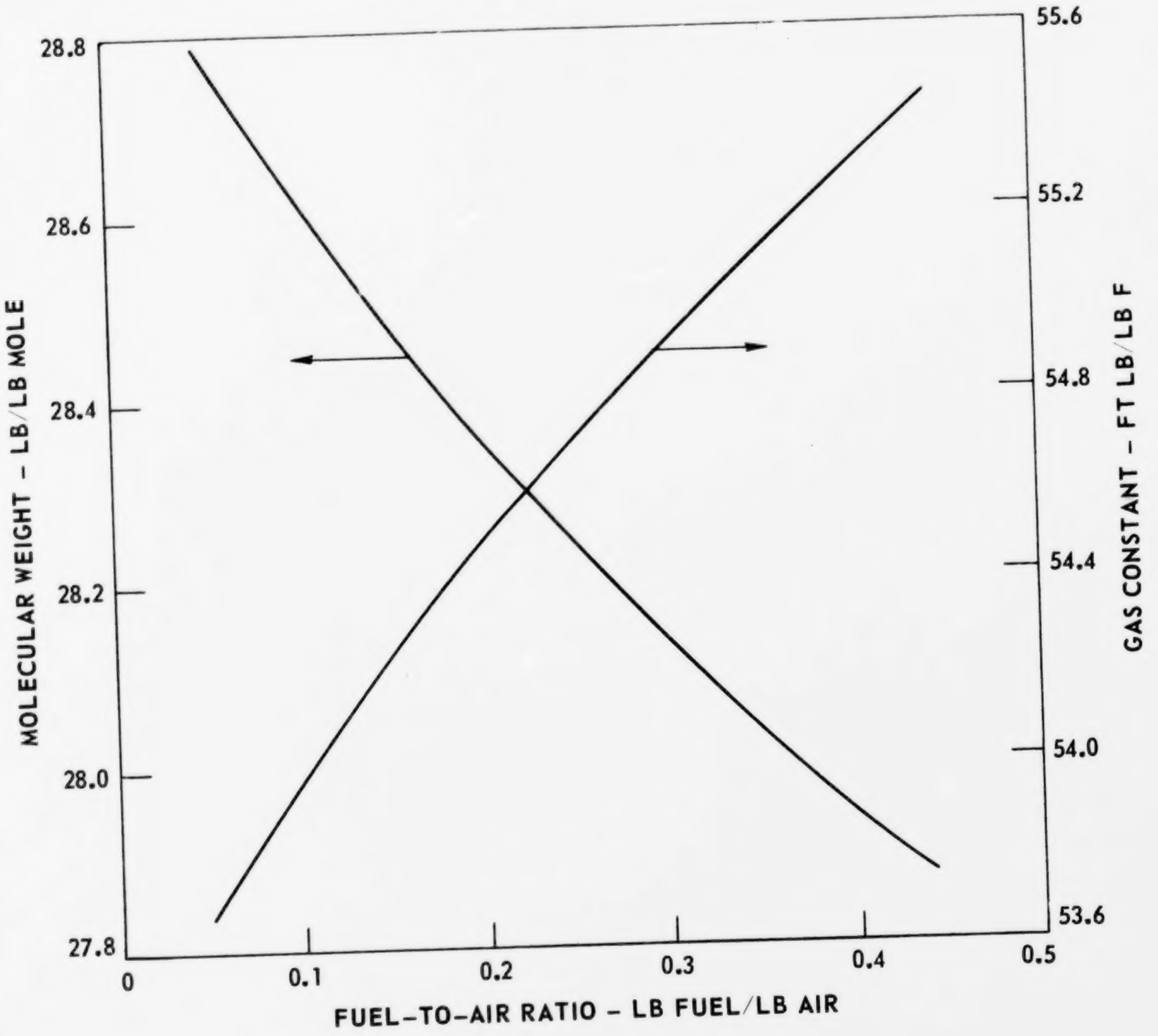


FIG. 165

MOLECULAR WEIGHT AND GAS CONSTANT OF PRODUCTS OF COMBUSTION OF FUEL GAS II



SPECIFIC HEAT AND SPECIFIC HEAT RATIO OF FUEL GAS III

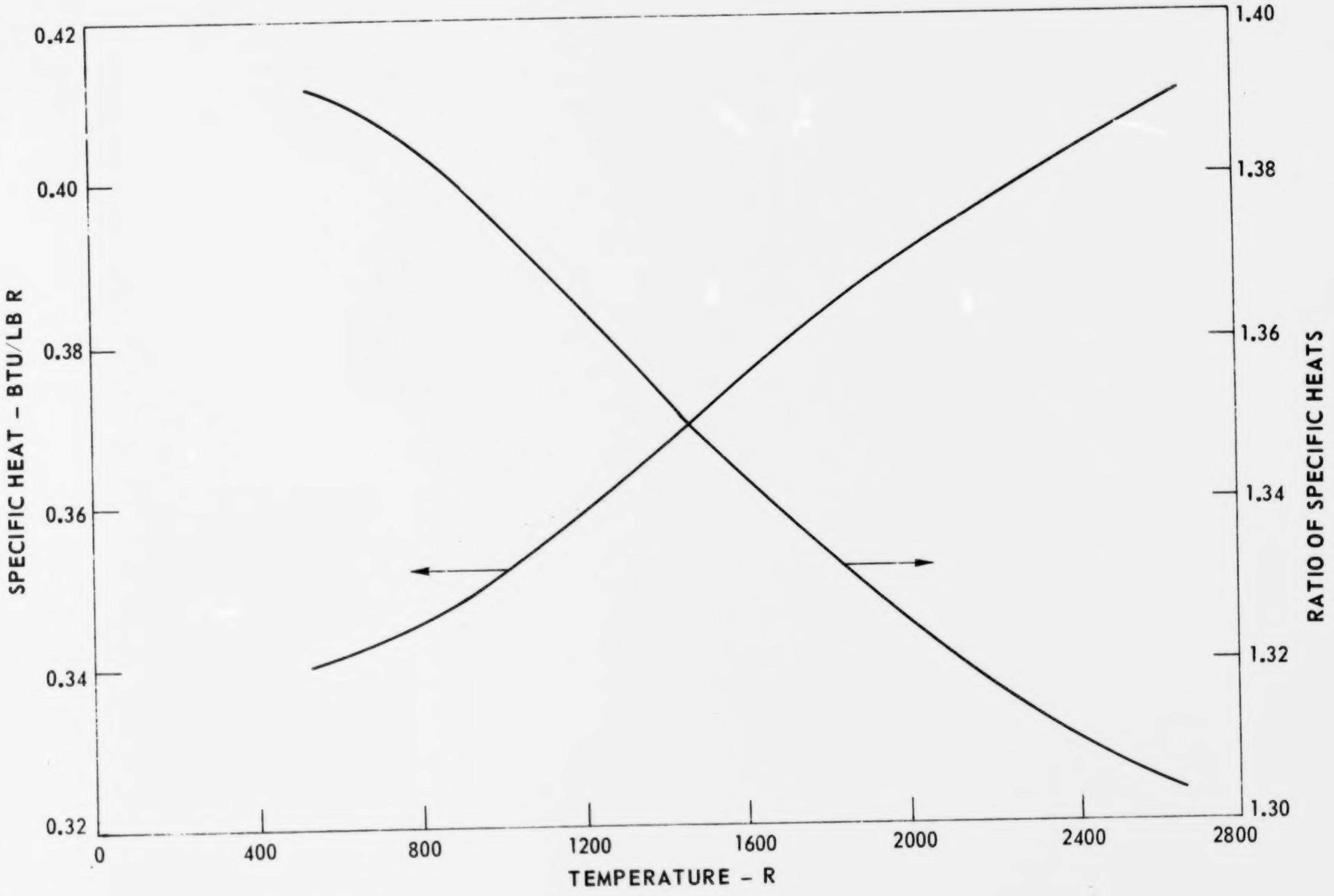


FIG. 166

FIG. 167

TEMPERATURE RISE OF FUEL GAS III

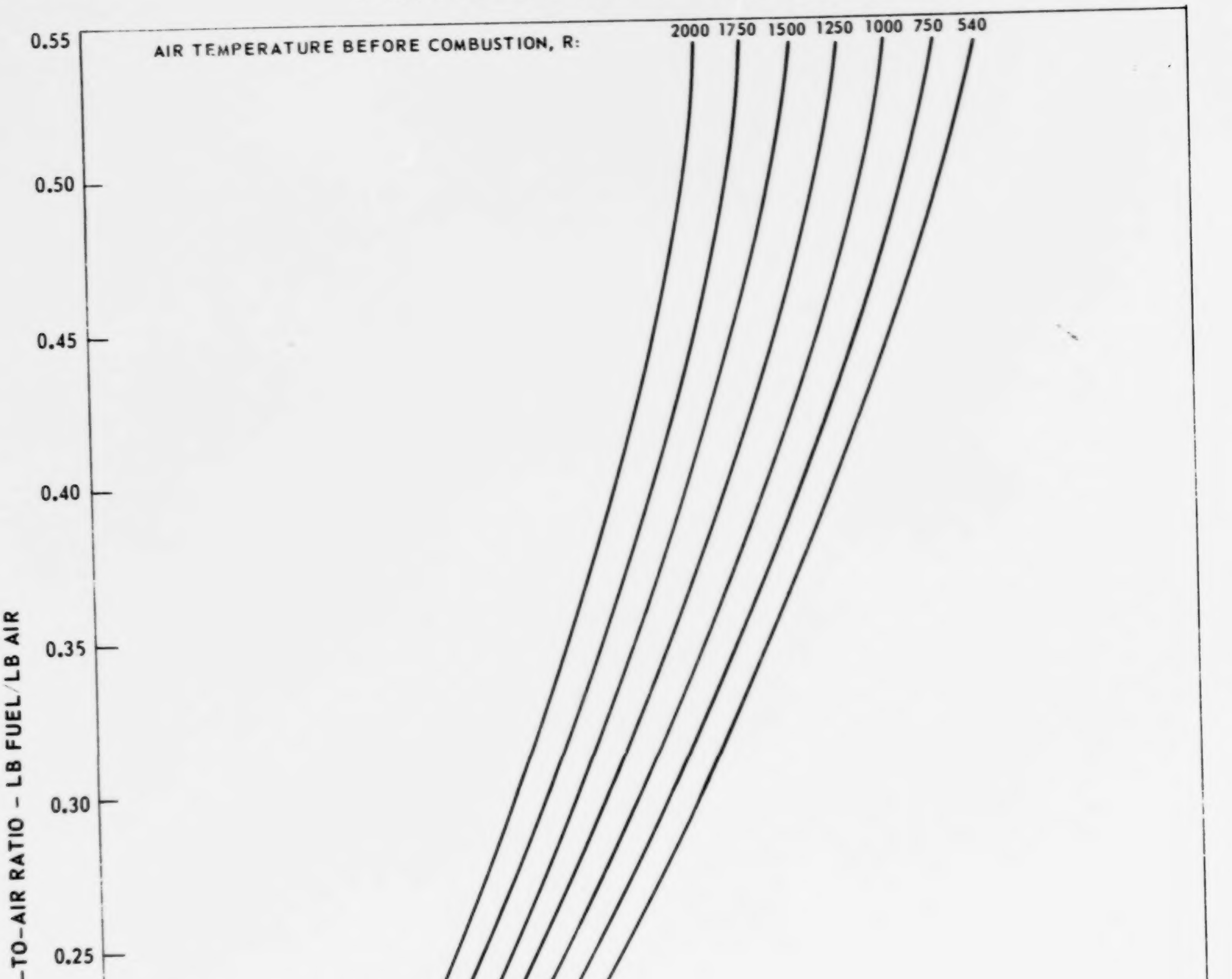
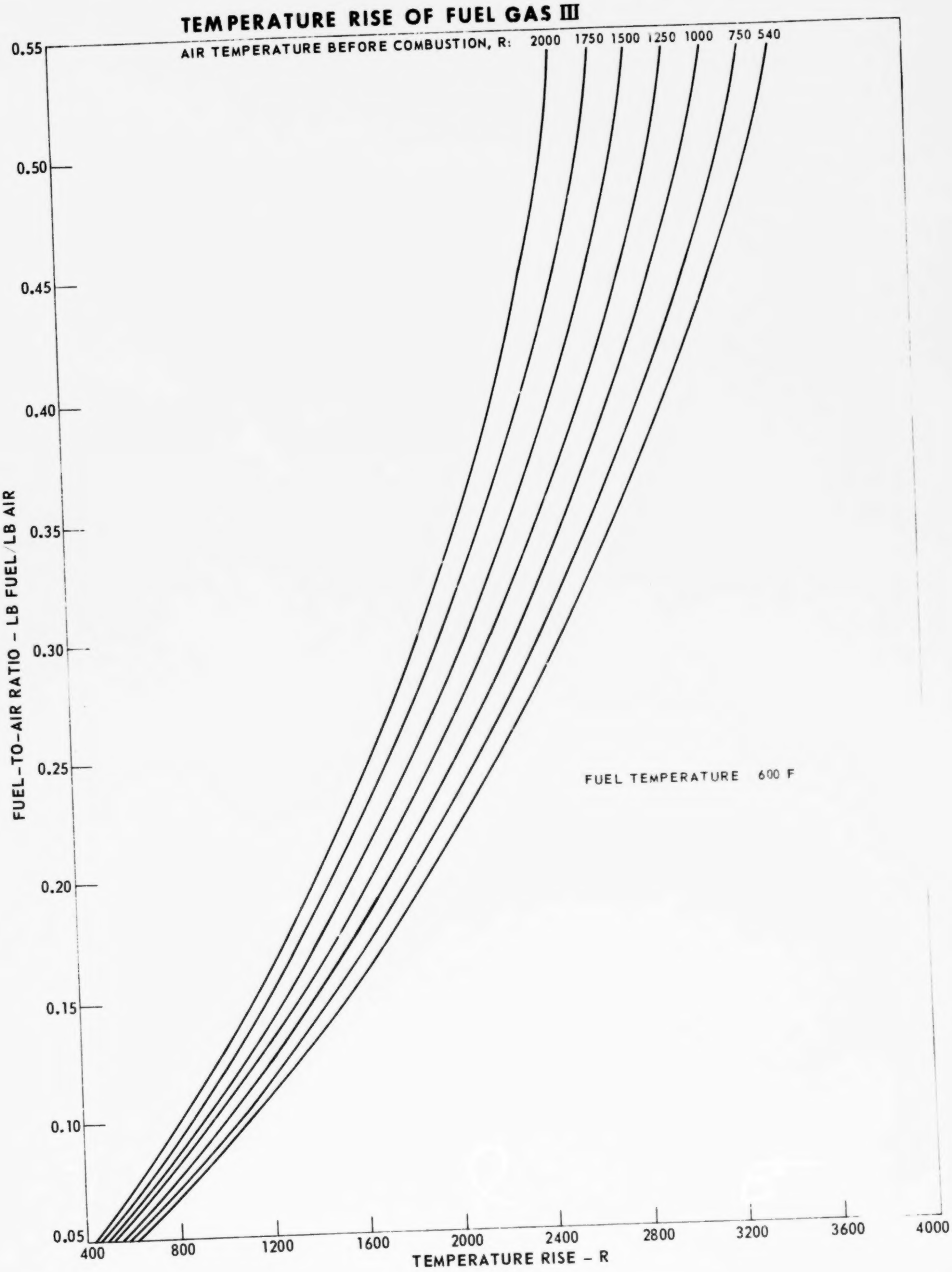


FIG. 168



TEMPERATURE RISE OF FUEL GAS III

FIG. 169

TEMPERATURE RISE OF FUEL GAS III

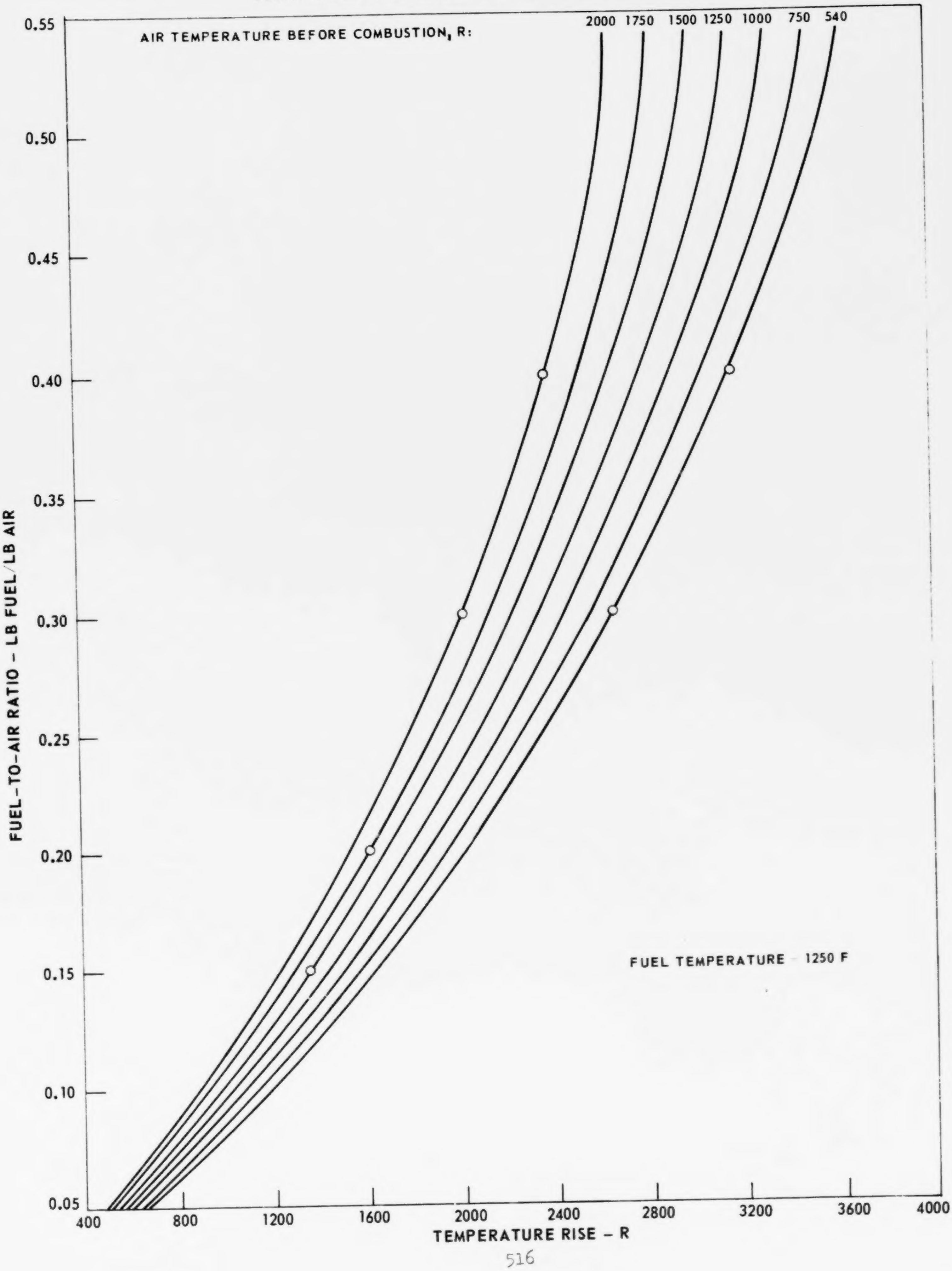
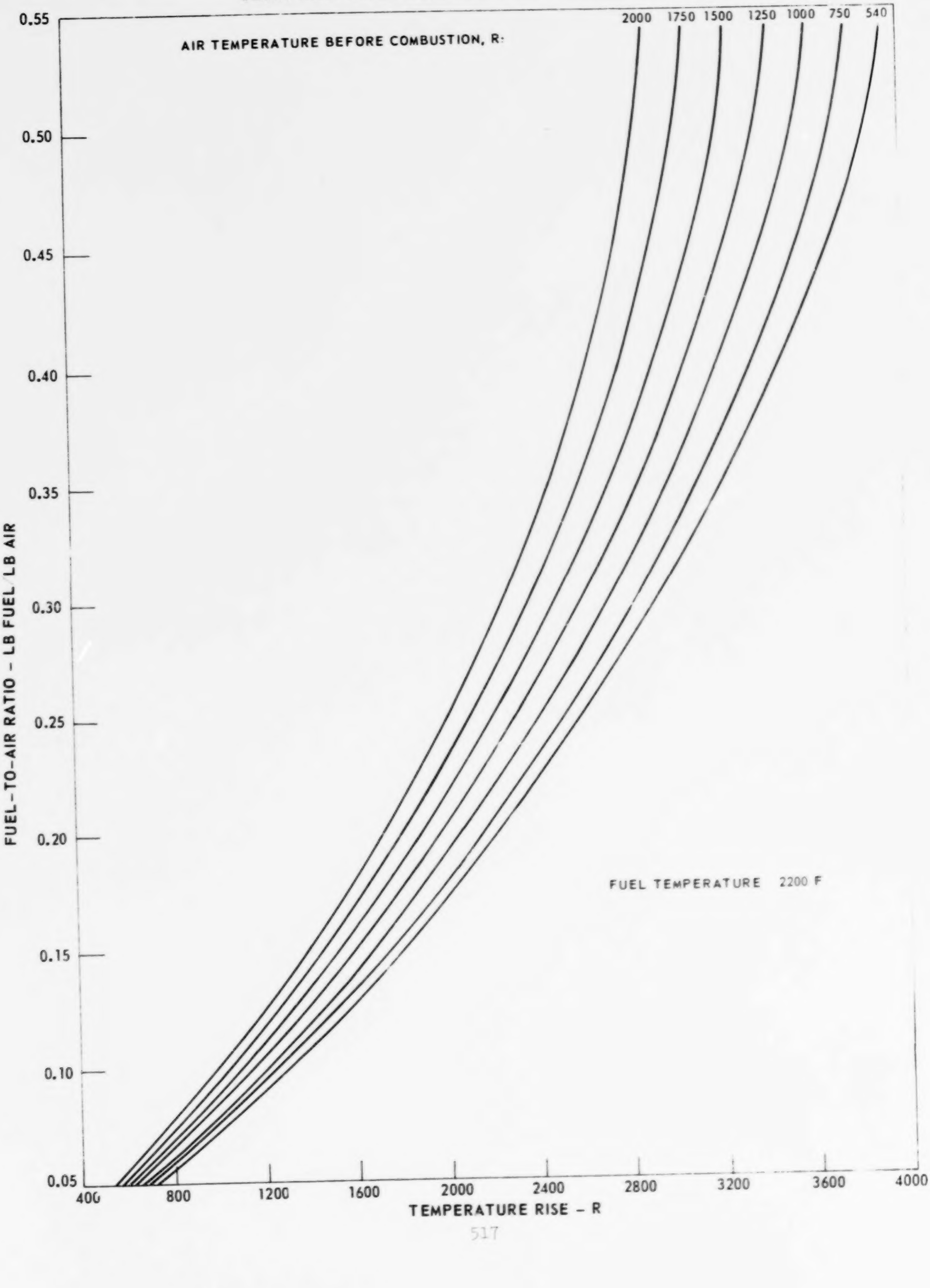


FIG. 170

TEMPERATURE RISE OF FUEL GAS III



TEMPERATURE RISE OF FUEL GAS III

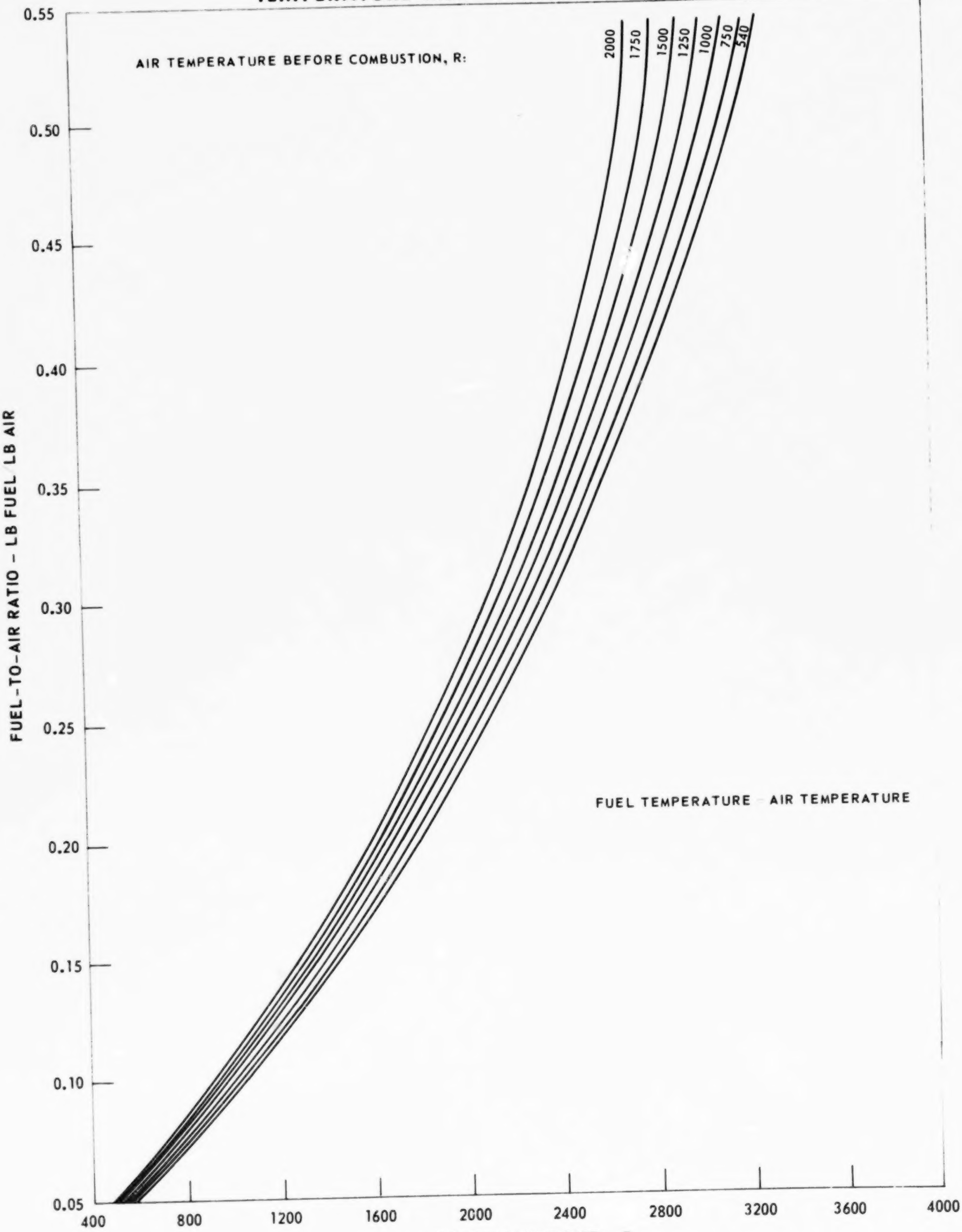


FIG. 172

SPECIFIC HEAT OF PRODUCTS OF COMBUSTION OF FUEL GAS III

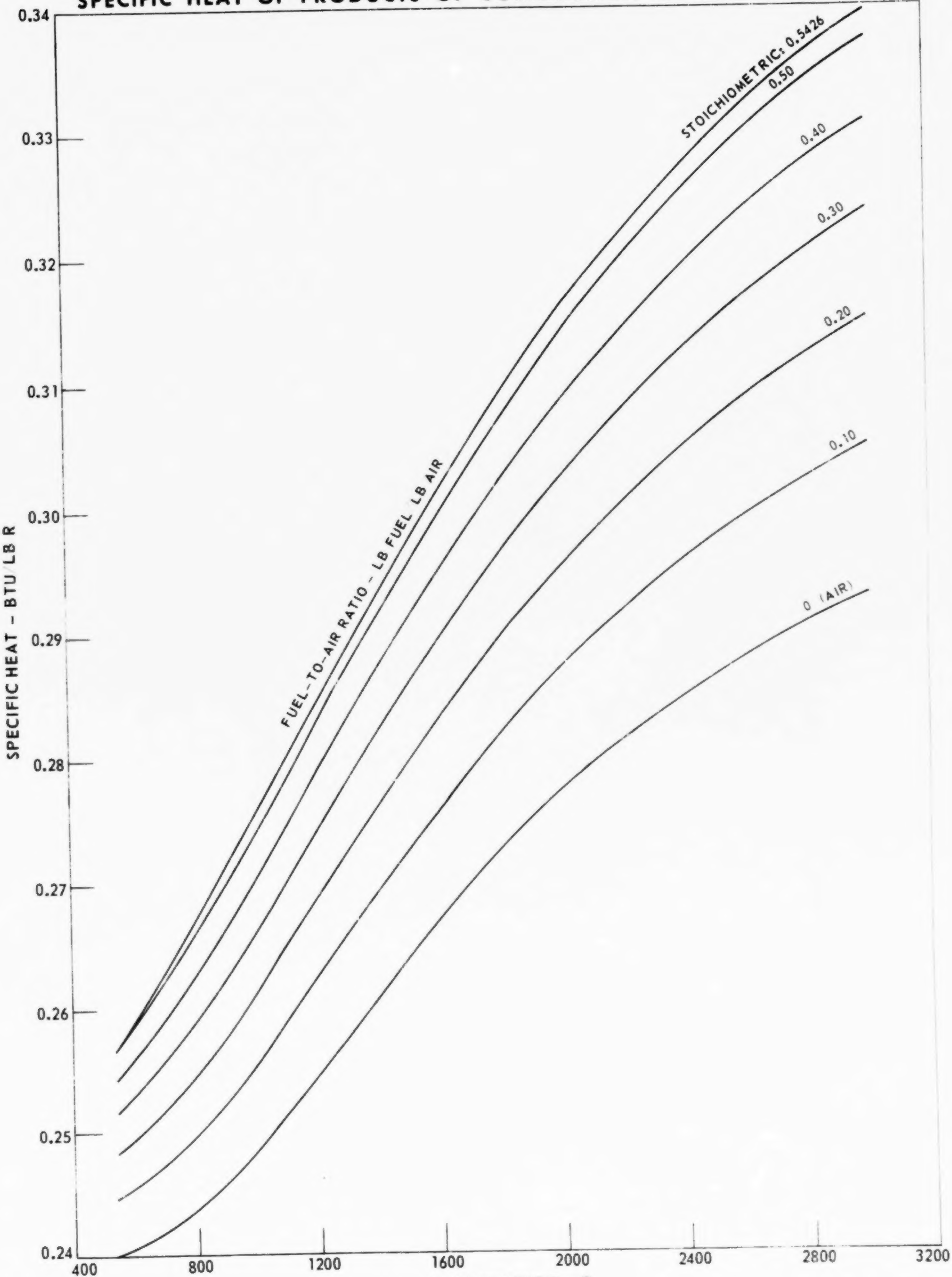
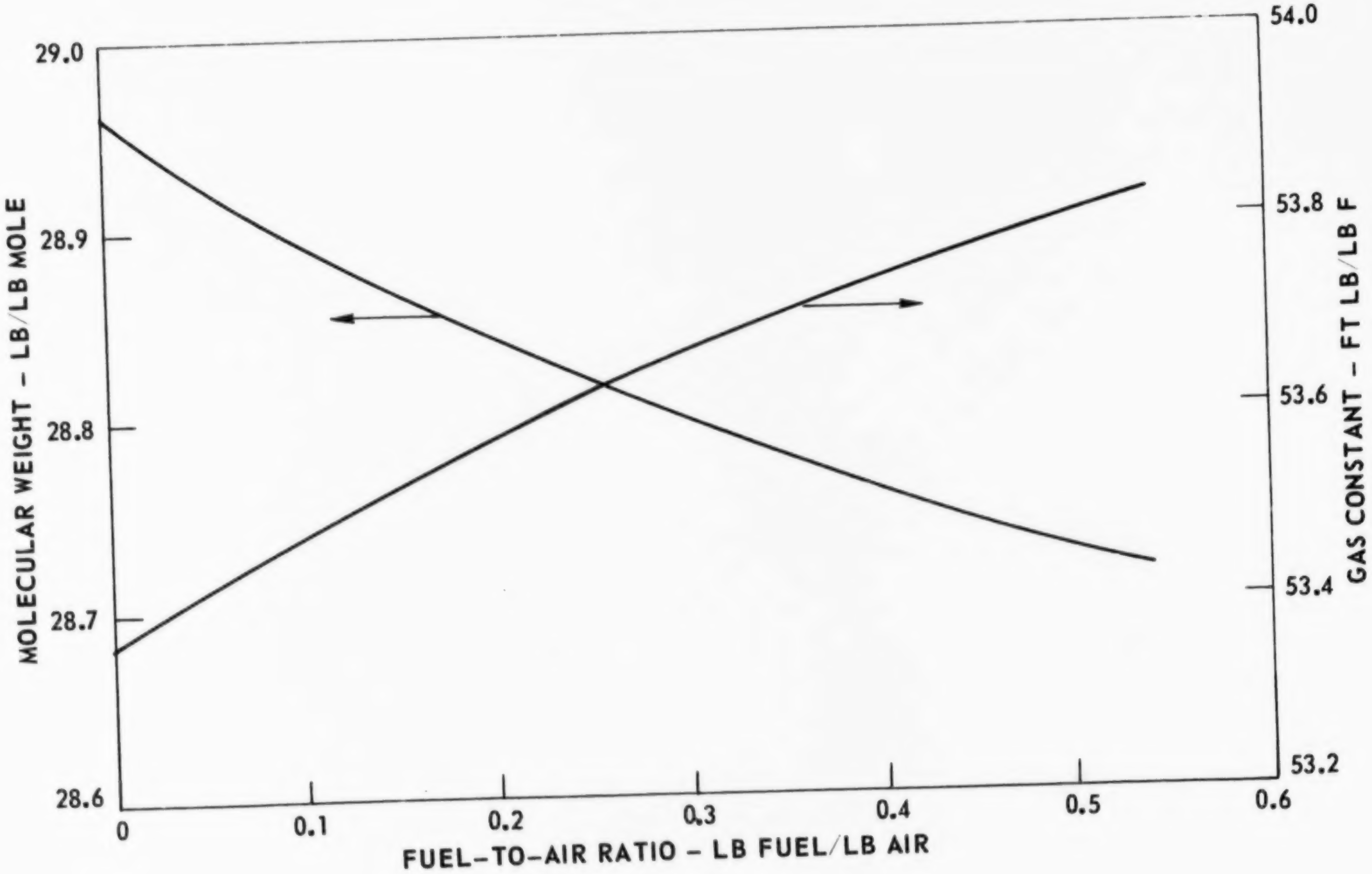


FIG. 173

MOLECULAR WEIGHT AND GAS CONSTANT OF PRODUCTS OF COMBUSTION OF FUEL GAS III



SPECIFIC HEAT AND SPECIFIC HEAT RATIO OF METHANE

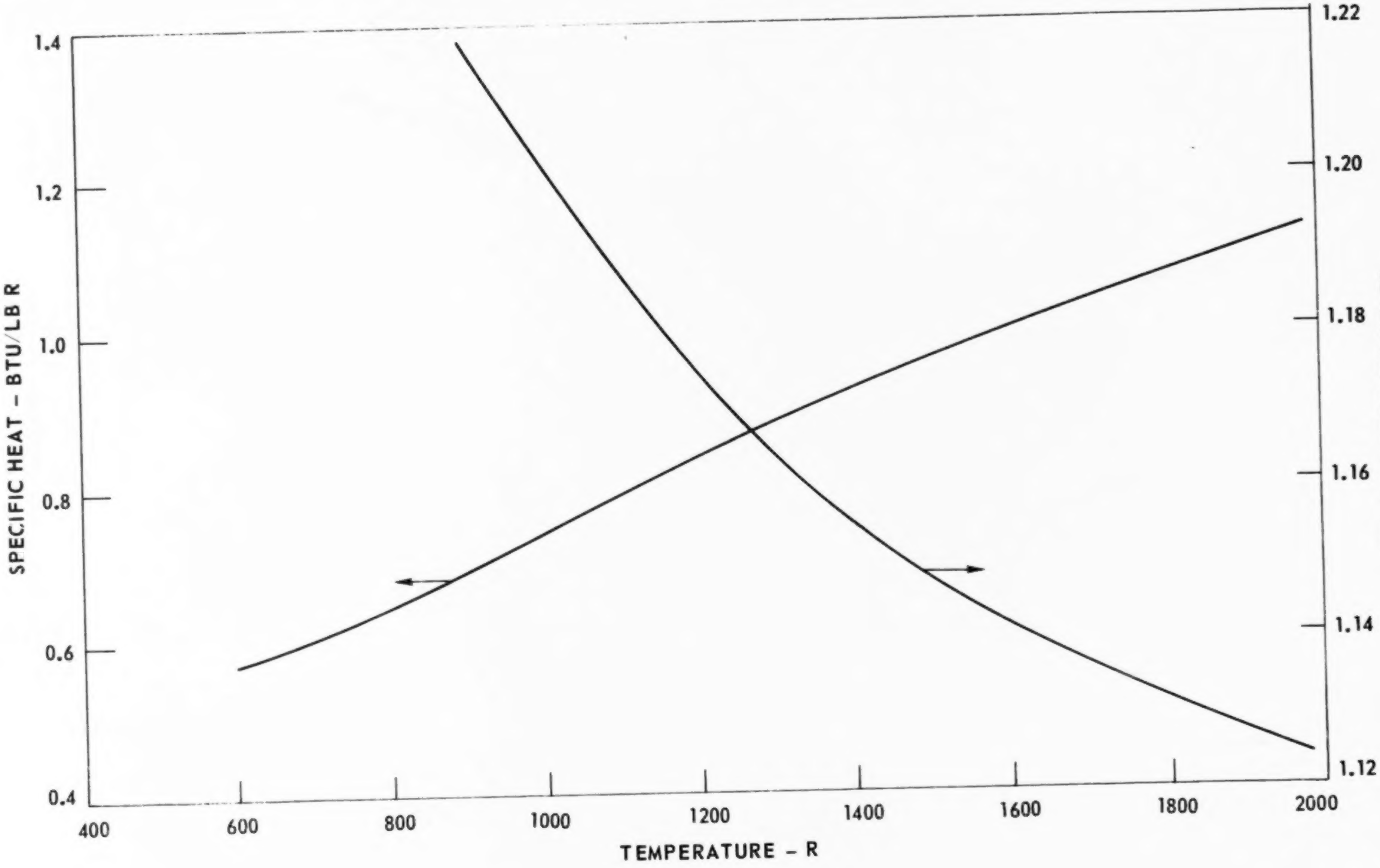


FIG. 174

FIG. 175

TEMPERATURE RISE OF METHANE

FUEL TEMPERATURE = 80 F

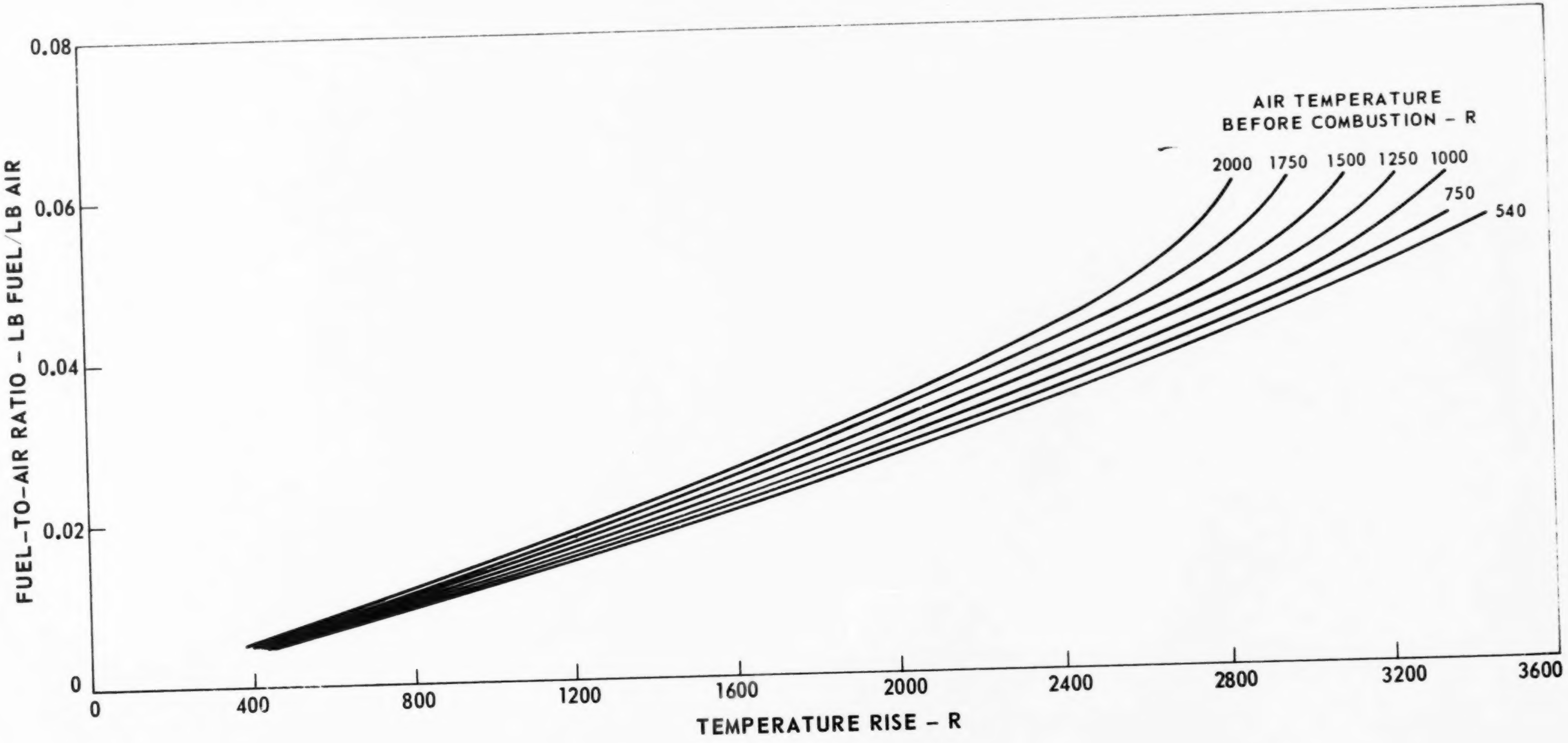


FIG. 176

SPECIFIC HEAT OF PRODUCTS OF COMBUSTION OF METHANE

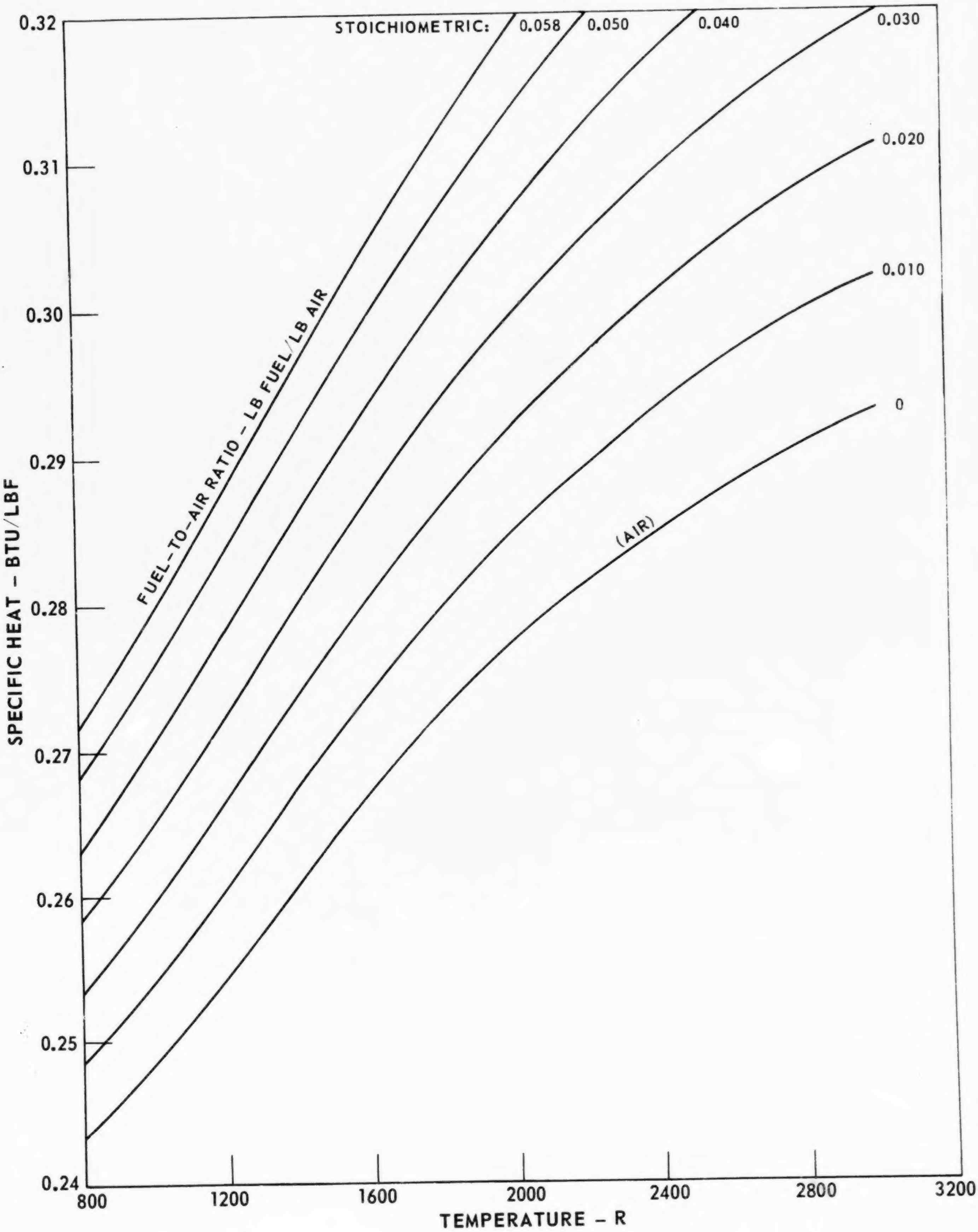
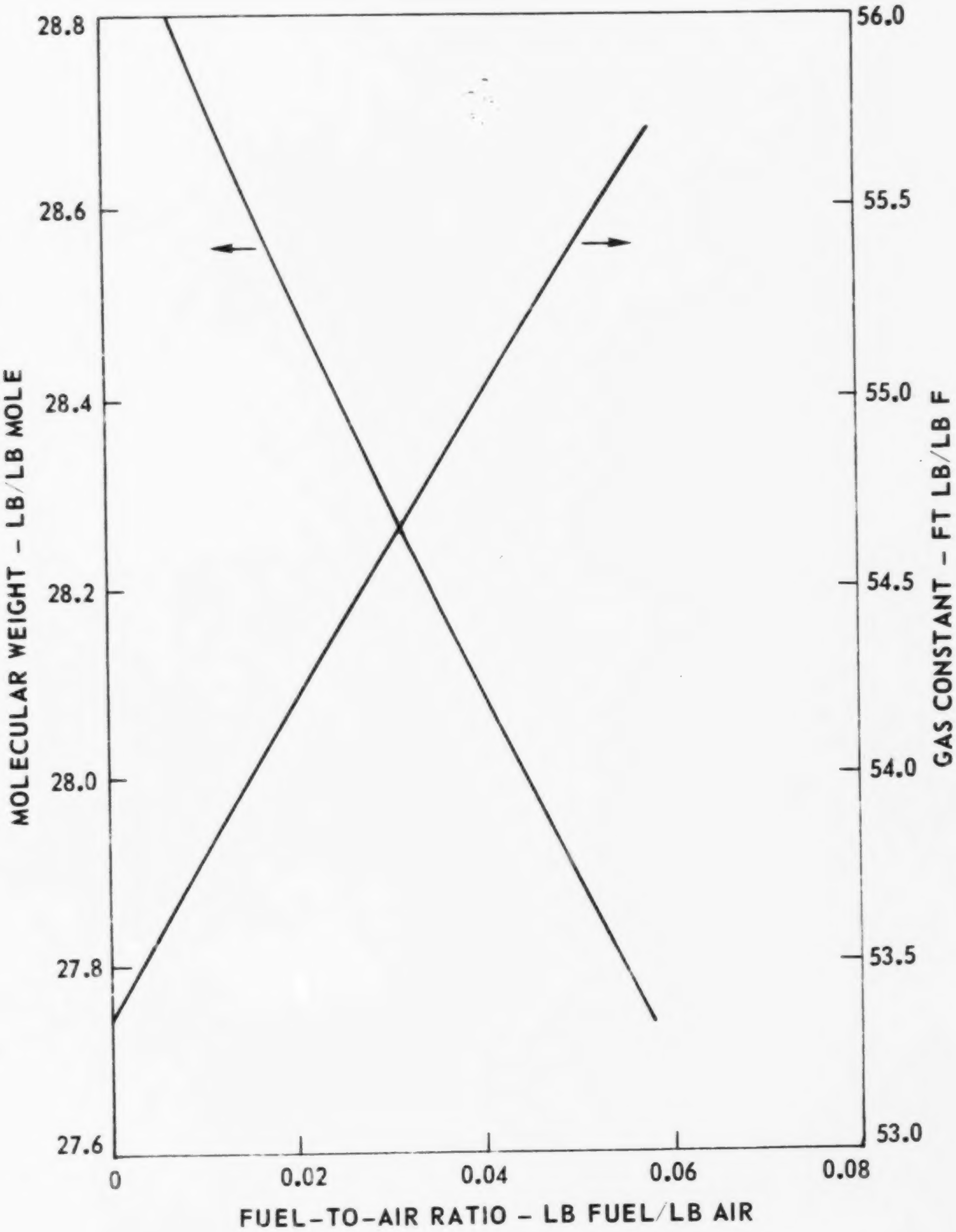


FIG. 177

MOLECULAR WEIGHT AND GAS CONSTANT OF PRODUCTS OF COMBUSTION OF METHANE



SCHEMATIC DIAGRAM OF MODEL USED IN GAS TURBINE COST ANALYSIS

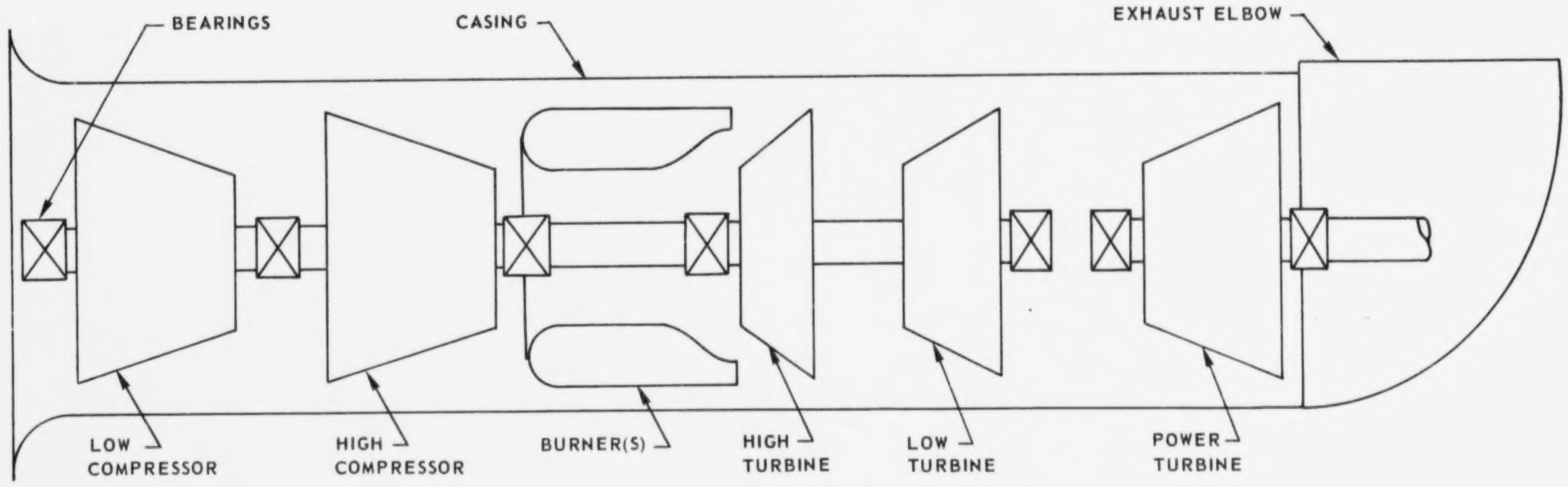


FIG. 178

FIG. 179

ILLUSTRATION OF TYPICAL COMPRESSOR BLADE DRAWINGS SUPPLIED TO BLADE MANUFACTURERS

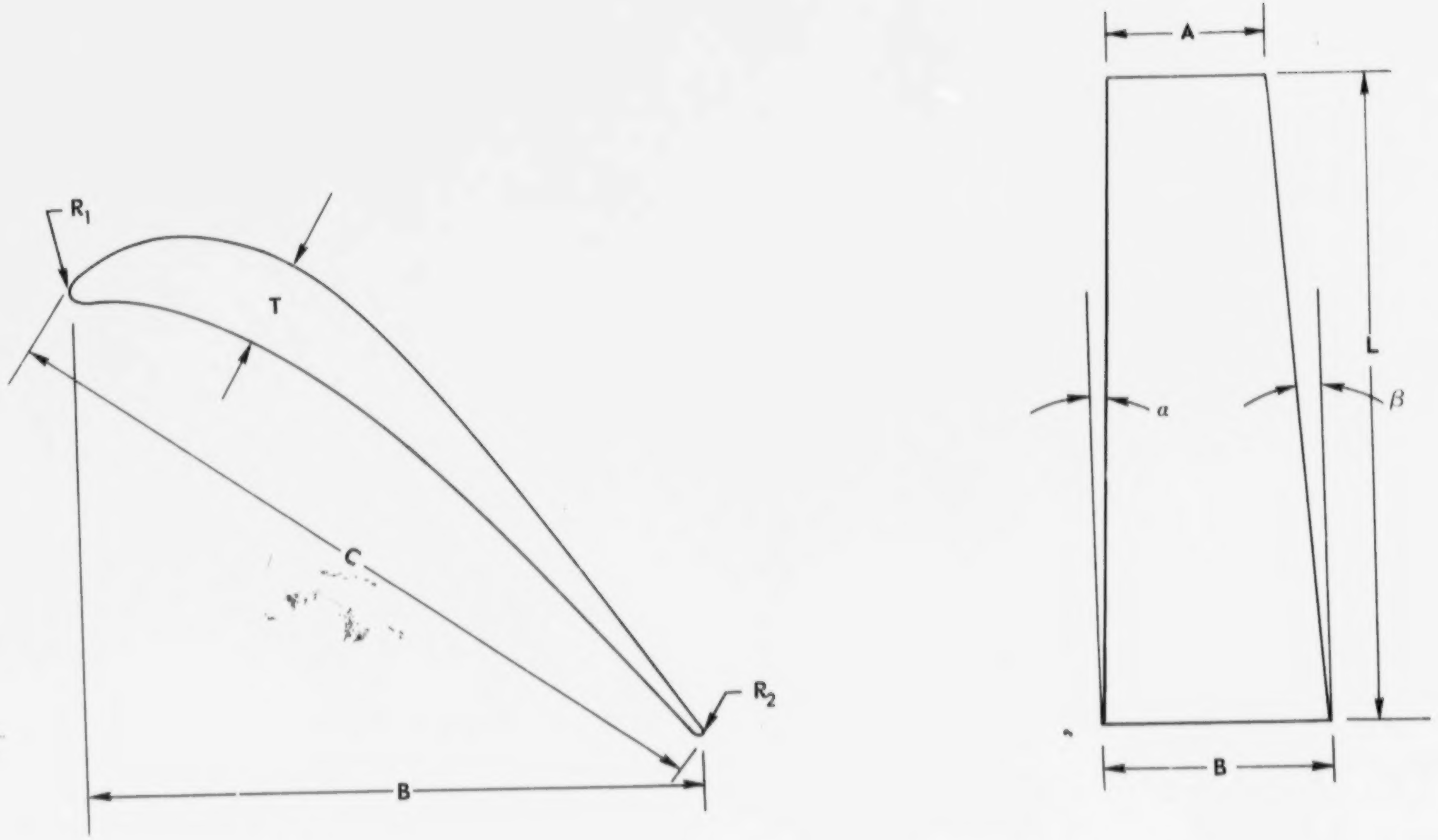
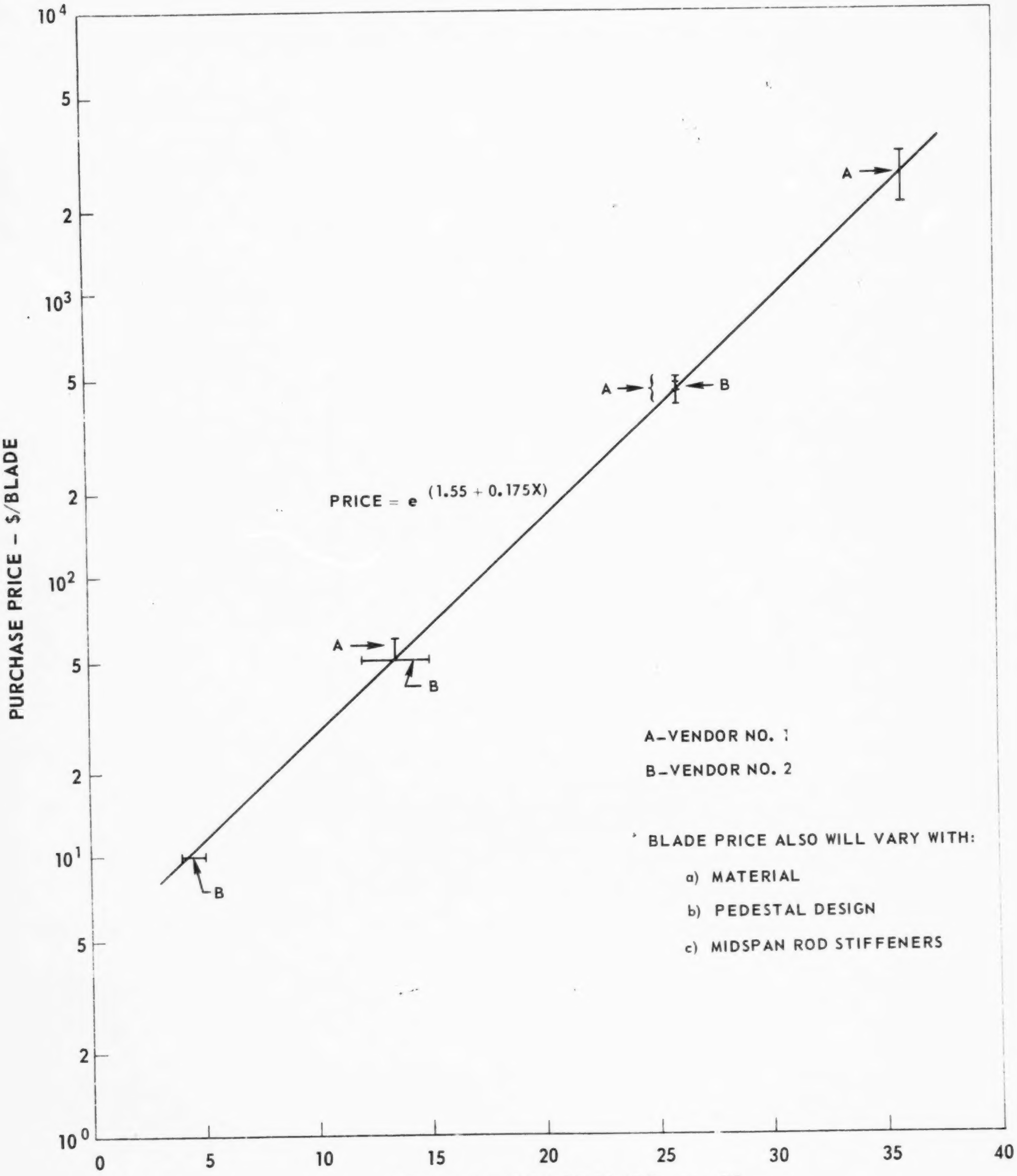


FIG. 180

MANUFACTURERS' PRICES FOR SOLID COMPRESSOR AND TURBINE BLADES

MATERIAL-AMS 4928
STRAIGHT PEDESTAL DESIGN



A-VENDOR NO. 1
B-VENDOR NO. 2

BLADE PRICE ALSO WILL VARY WITH:
a) MATERIAL
b) PEDESTAL DESIGN
c) MIDSPAN ROD STIFFENERS

FIG. 181

ILLUSTRATION OF A TYPICAL COMPRESSOR DISK SUPPLIED TO DISK MANUFACTURER

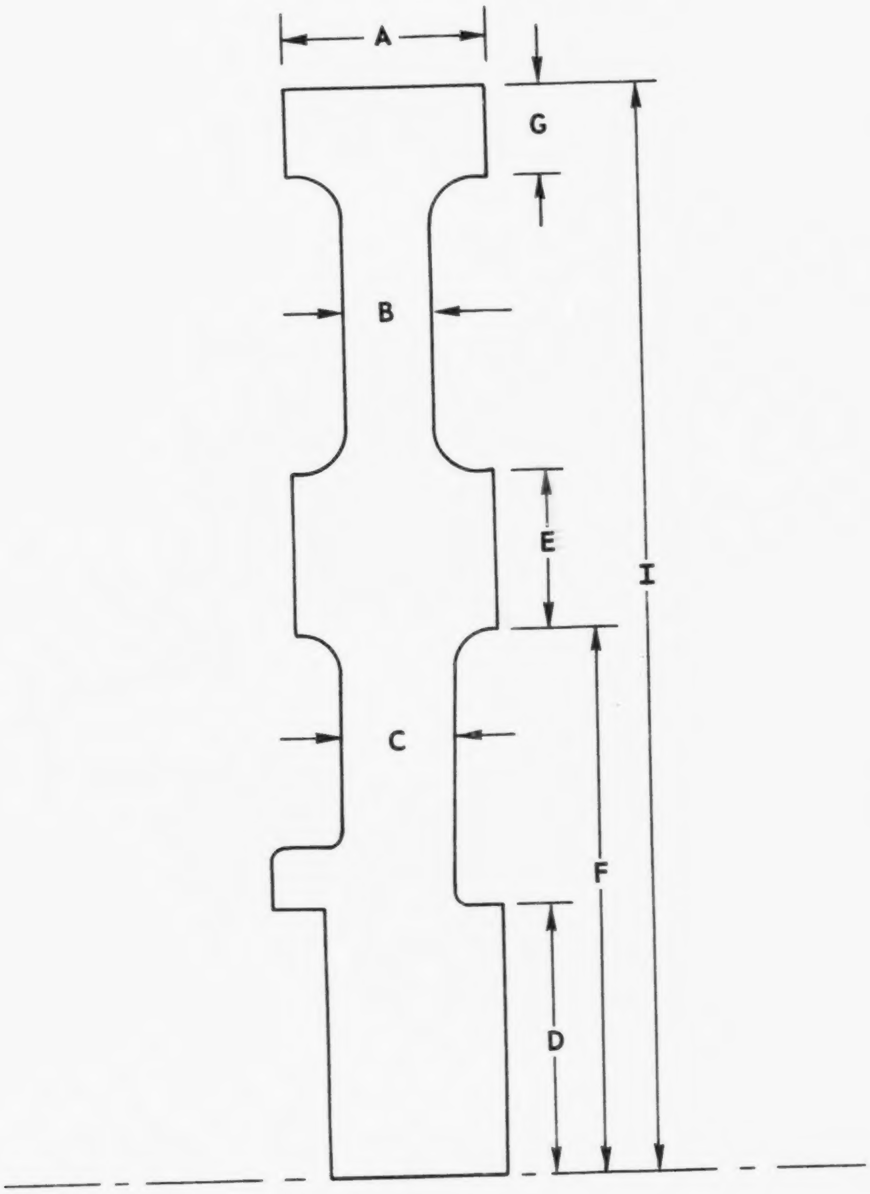


FIG. 182

PRICE ESTIMATES FOR FORGED COMPRESSOR DISKS

DATA SOURCE–VENDOR NO. 3
NO CORRECTION FOR PRODUCTION VOLUME

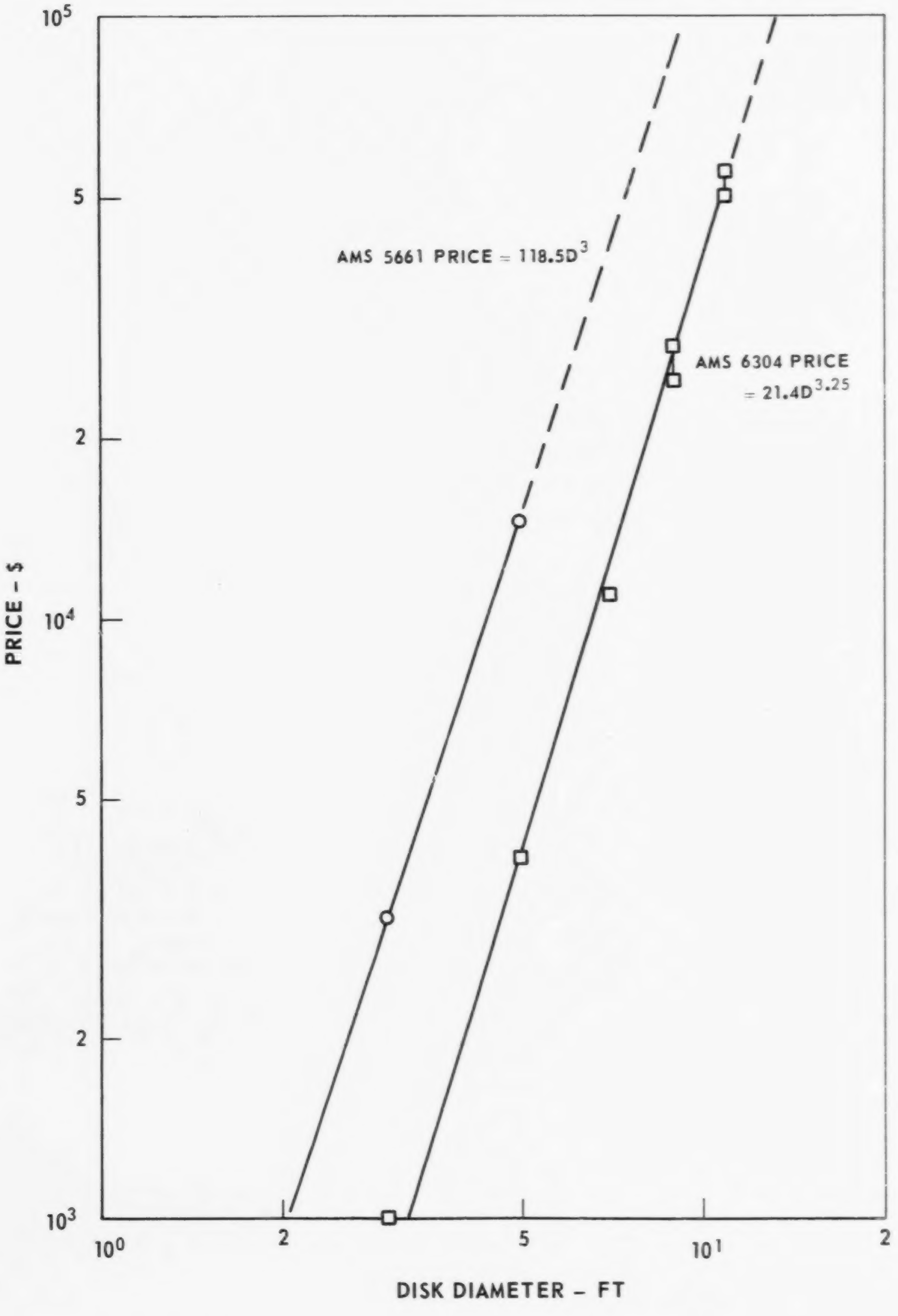


FIG. 183

TOOLING COSTS FOR TURBINE AND COMPRESSOR DISKS

DATA SOURCE: VENDOR NO. 3

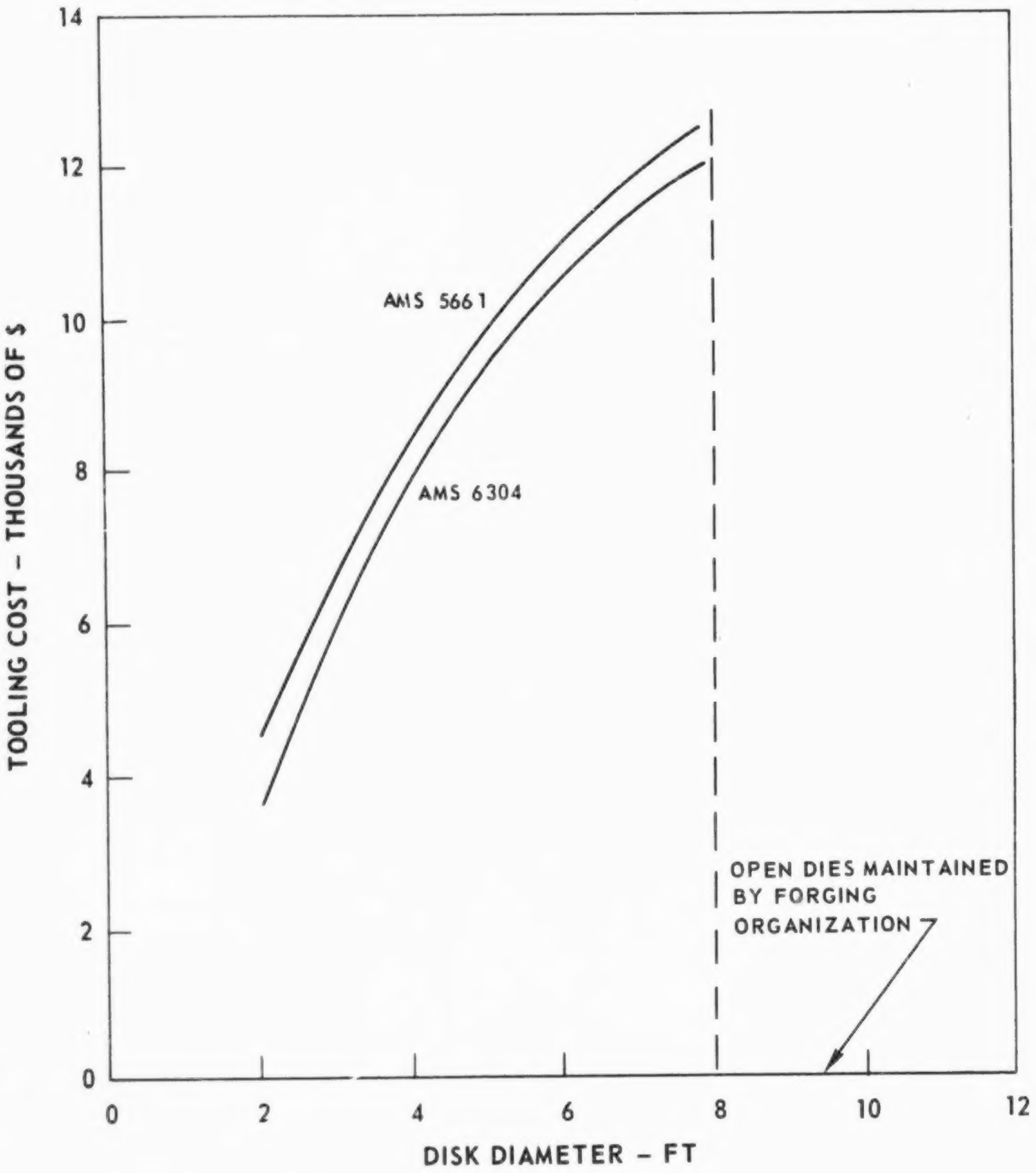


ILLUSTRATION OF TYPICAL IMPINGEMENT-COOLED TURBINE BLADE DRAWINGS SENT TO BLADE MANUFACTURERS

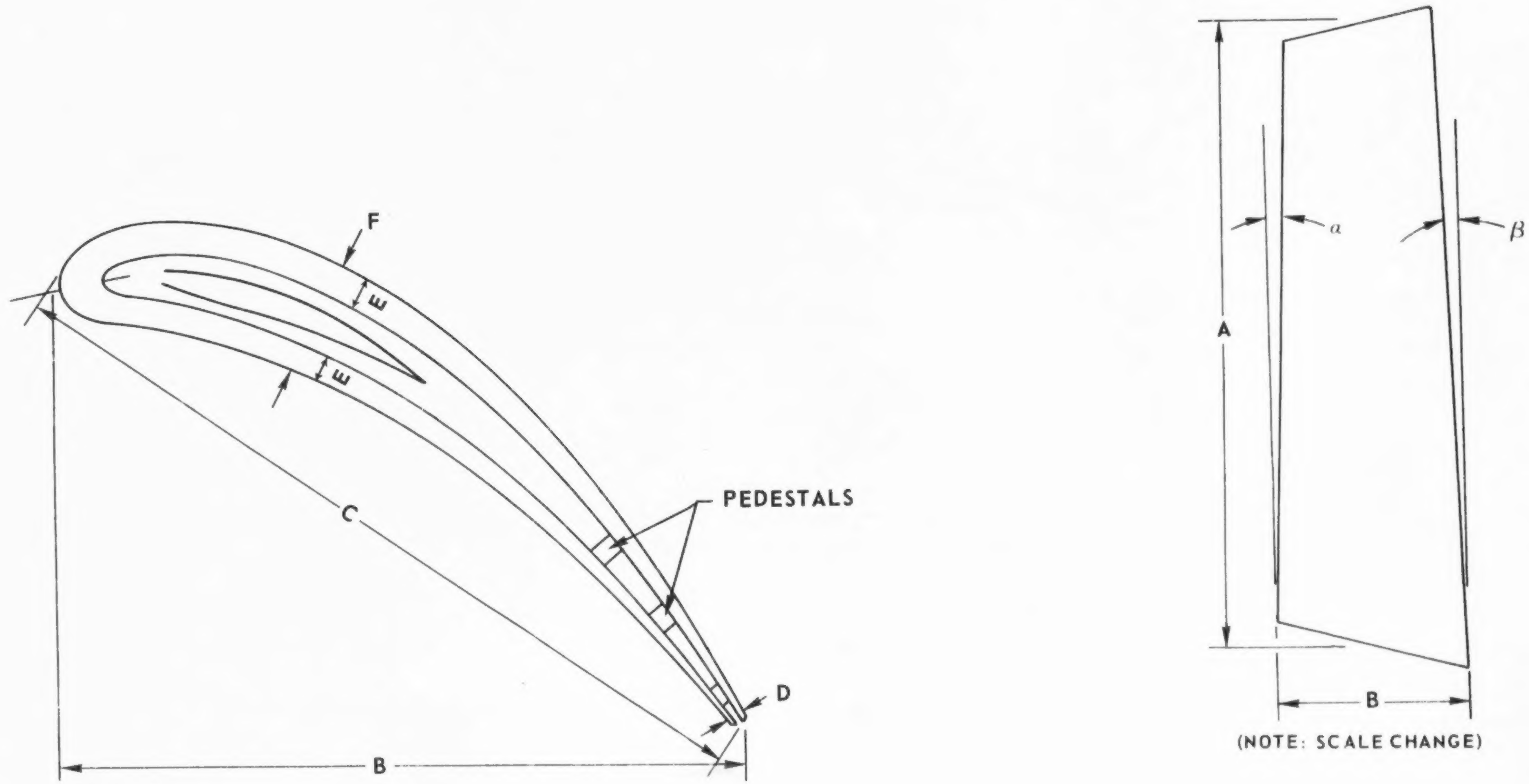


FIG. 184

FIG. 185

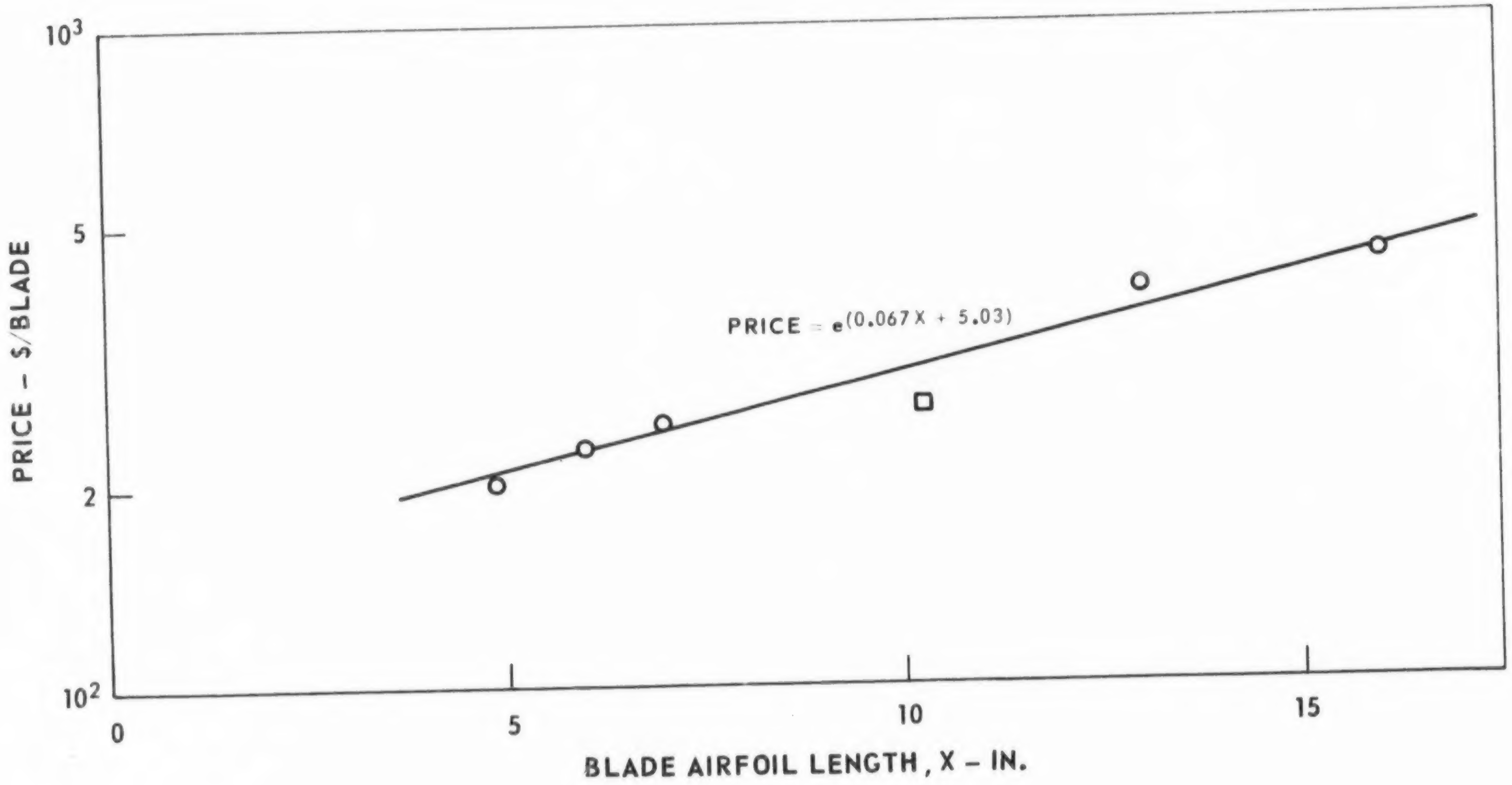
MANUFACTURERS' PRICES FOR IMPINGEMENT-COOLED TURBINE BLADES

MATERIAL-B-1900

AMS 5661

DATA SOURCE: VENDORS NO. 2 AND NO. 4

TOOLING COSTS NOT INCLUDED



MANUFACTURERS' PRICES FOR IMPINGEMENT-COOLED TURBINE VANES

DATA SOURCE: VENDOR NO. 2

TOOLING COSTS NOT INCLUDED

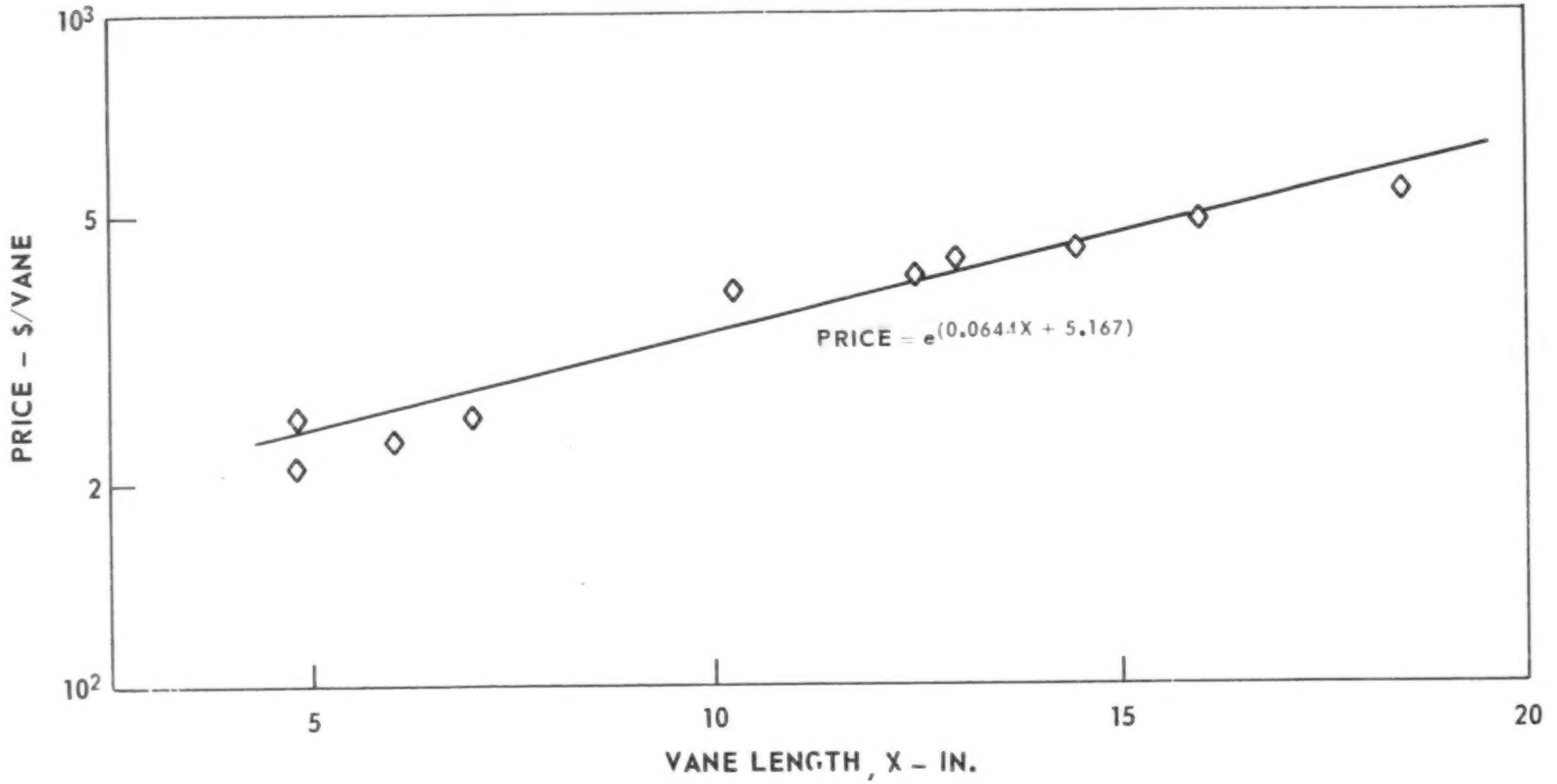


FIG. 186

FIG. 187

DRAWING OF TURBINE DISK SENT TO DISK MANUFACTURER

SURFACES MARKED BY V
TO BE MACHINED BY PURCHASER

DIMENSIONS SHOWN ARE APPROXIMATE
AND FOR THE PURPOSE OF OBTAINING
BUDGETARY COST ESTIMATES ONLY

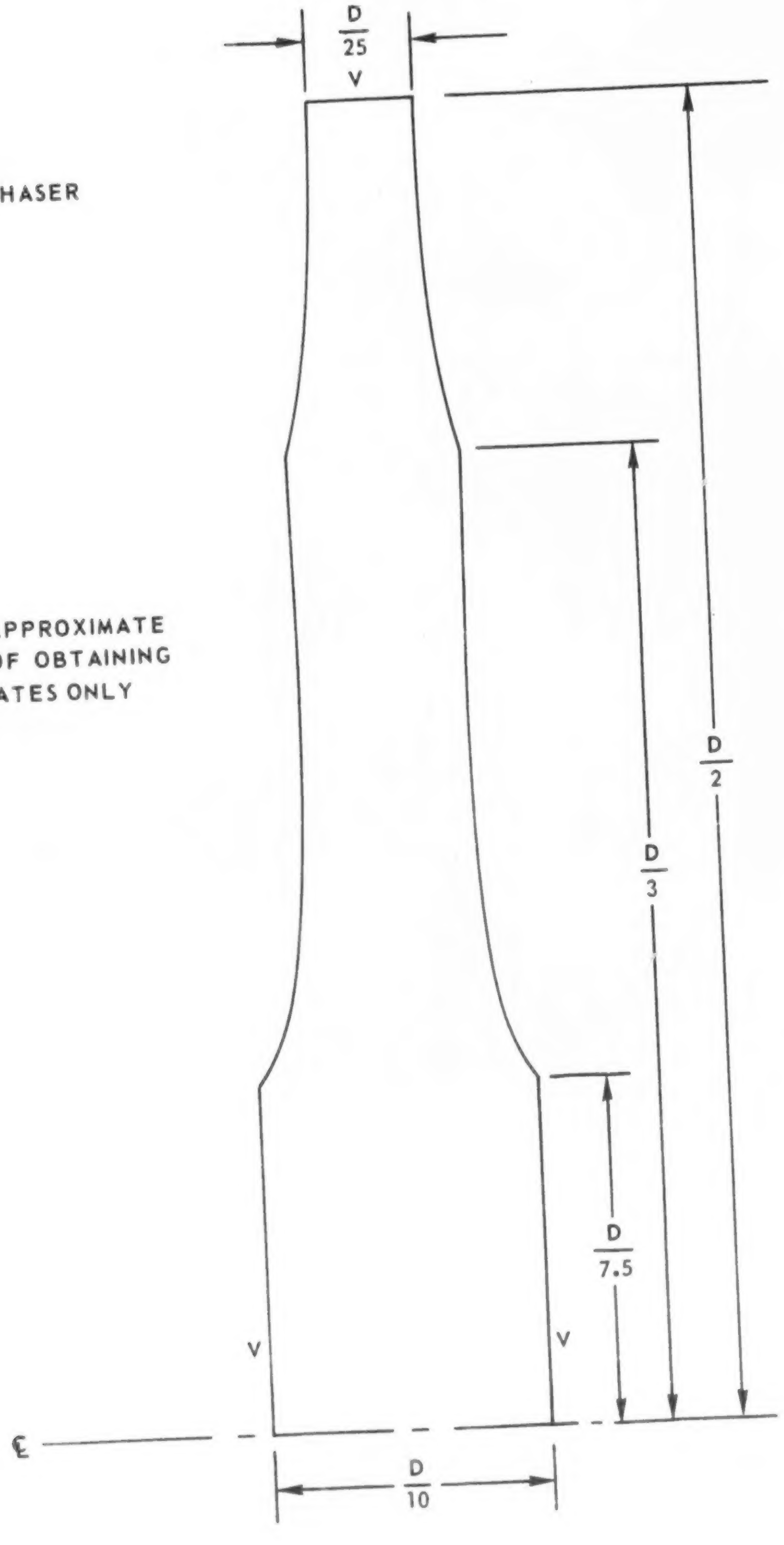


FIG. 188

PRICE ESTIMATES FOR FORGED TURBINE DISKS

DATA SOURCE: VENDOR NO. 3

NO CORRECTIONS FOR PRODUCTION VOLUME

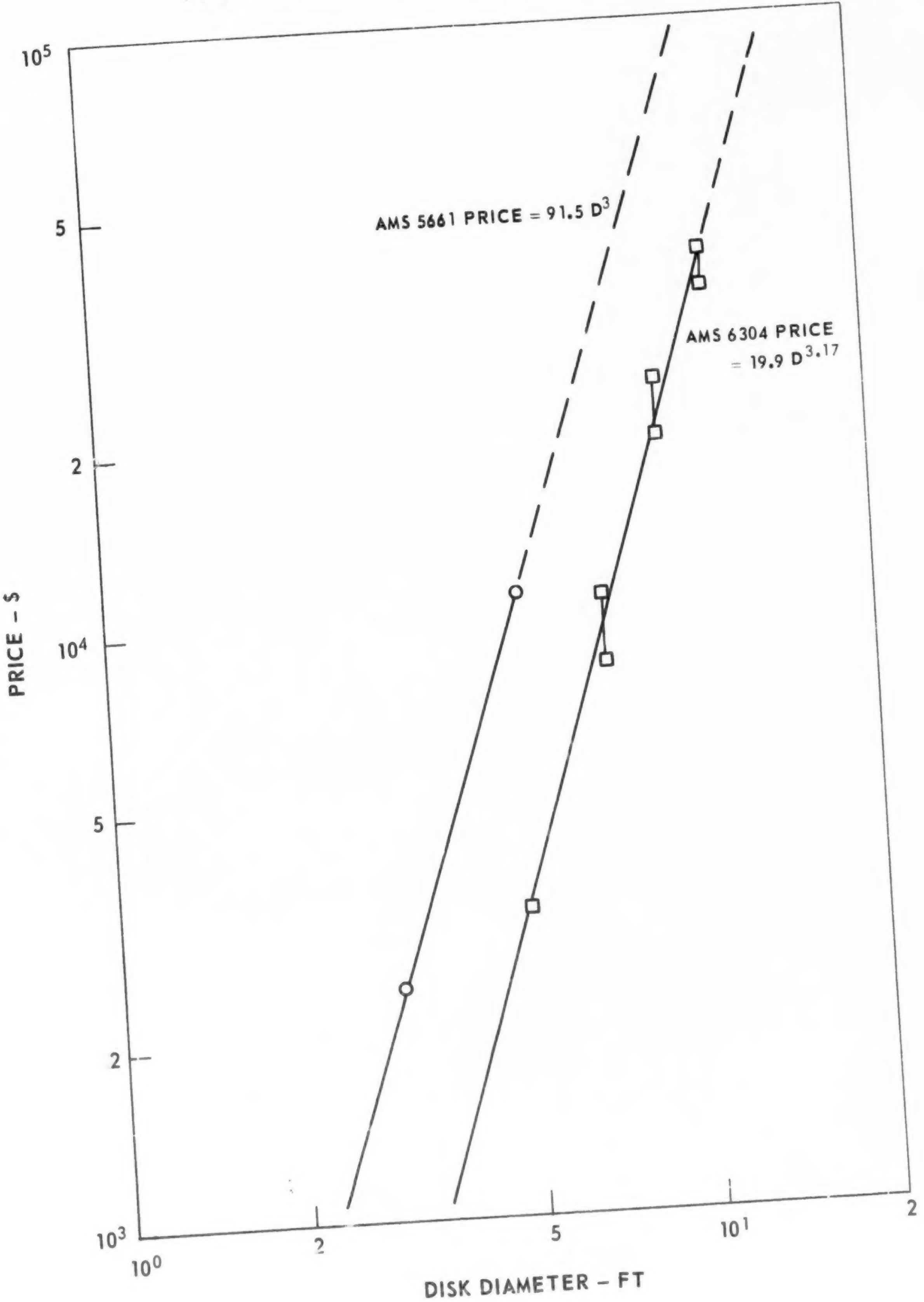
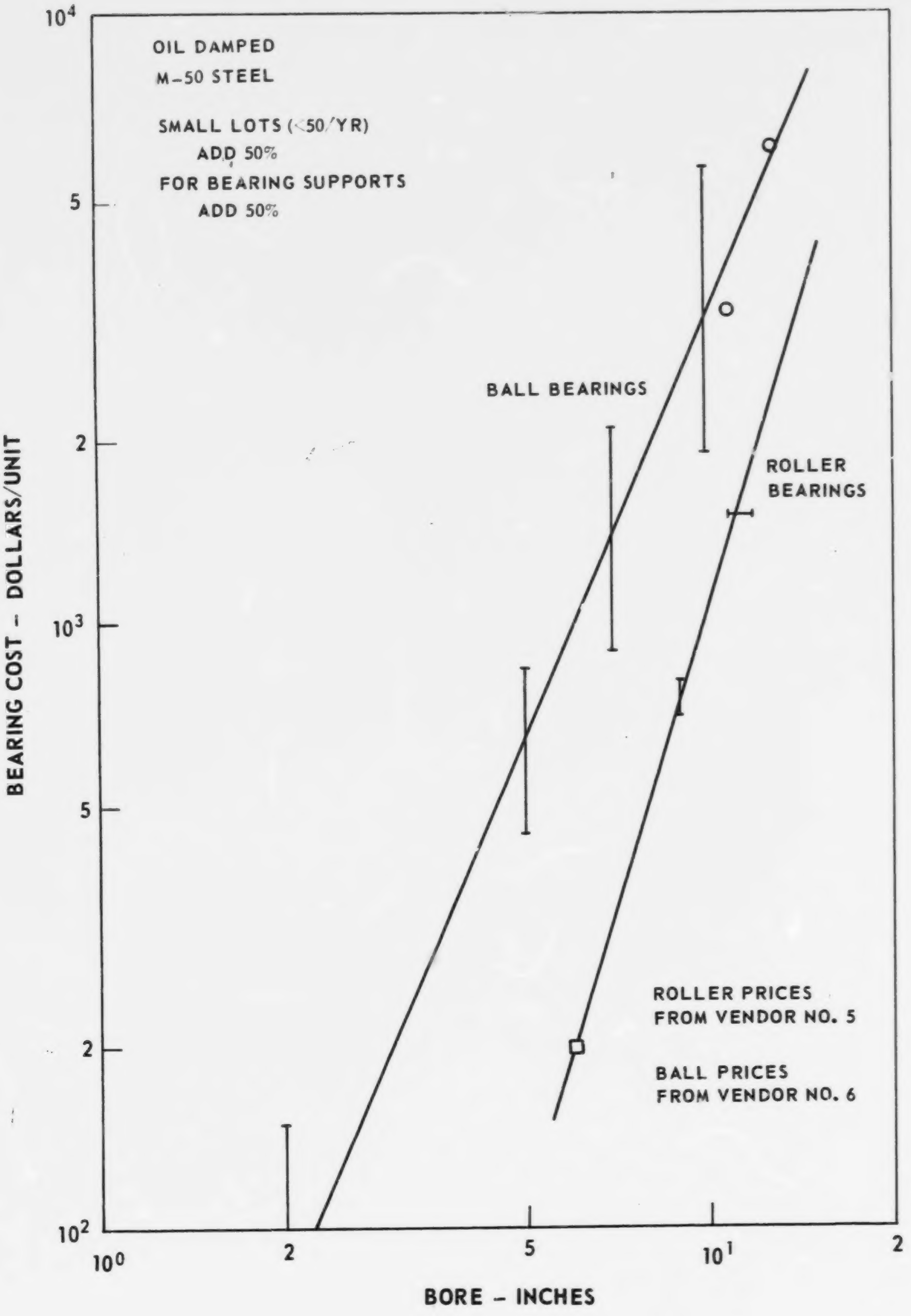
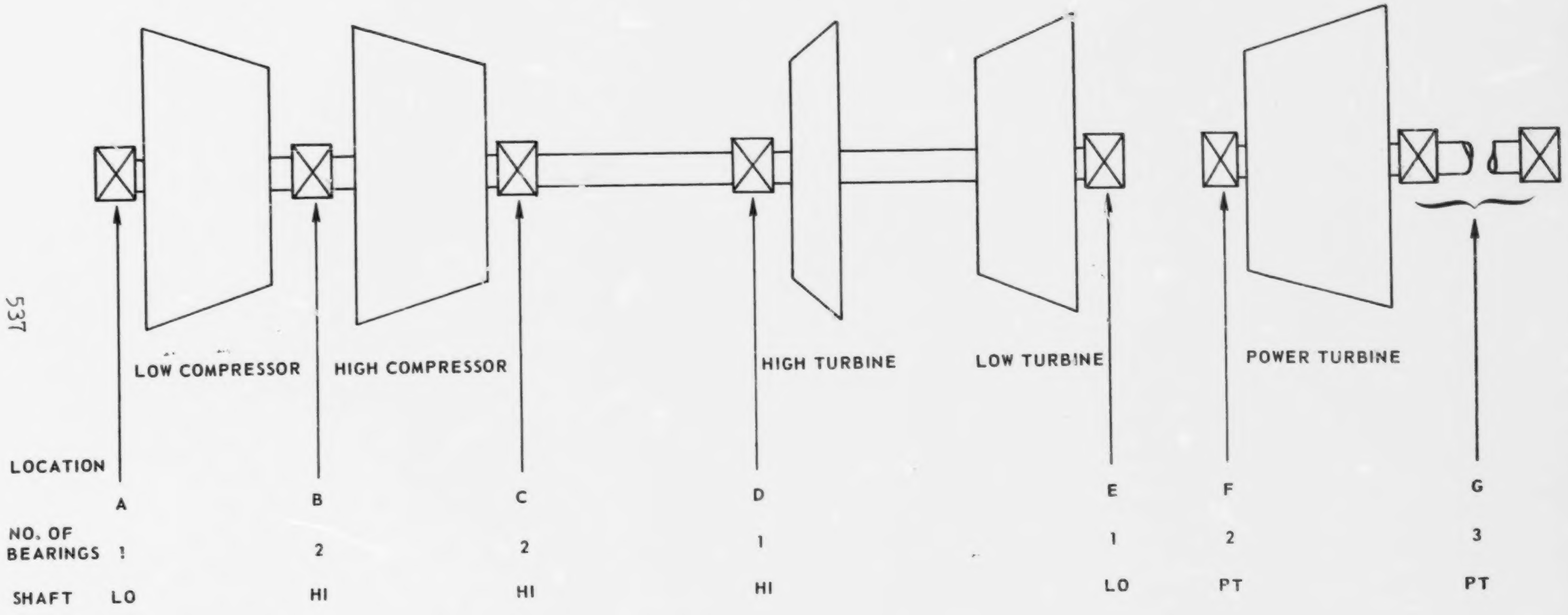


FIG. 189

SELLING PRICES FOR ROLLING-CONTACT BEARINGS



SCHEMATIC DIAGRAM OF ENGINE BEARING LOCATIONS



SHAFT DESIGNATION

HI–HIGH PRESSURE SECTION
LO–LOW PRESSURE SECTION
PT–POWER TURBINE SECTION

FIG. 190

FIG. 191

ESTIMATED SELLING PRICES AND MANUFACTURING COSTS OF SECOND-GENERATION GAS TURBINE ENGINES

TURBINE INLET TEMPERATURE – 2400 F

COMPRESSOR PRESSURE RATIO – 12

ANNUAL SALES – 10,000 MW

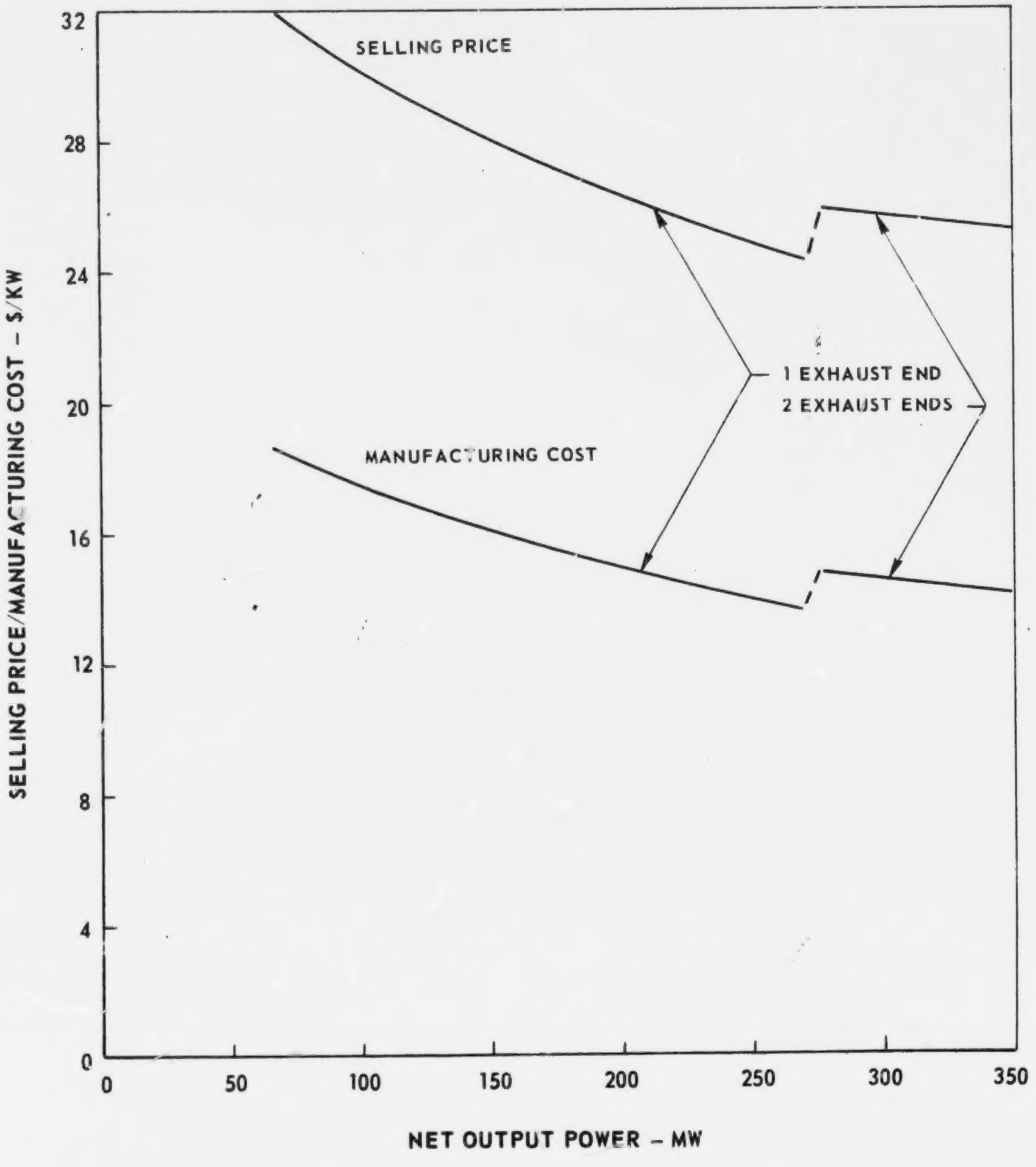


FIG. 192

THERMODYNAMIC PROPERTIES OF SULFUR DIOXIDE

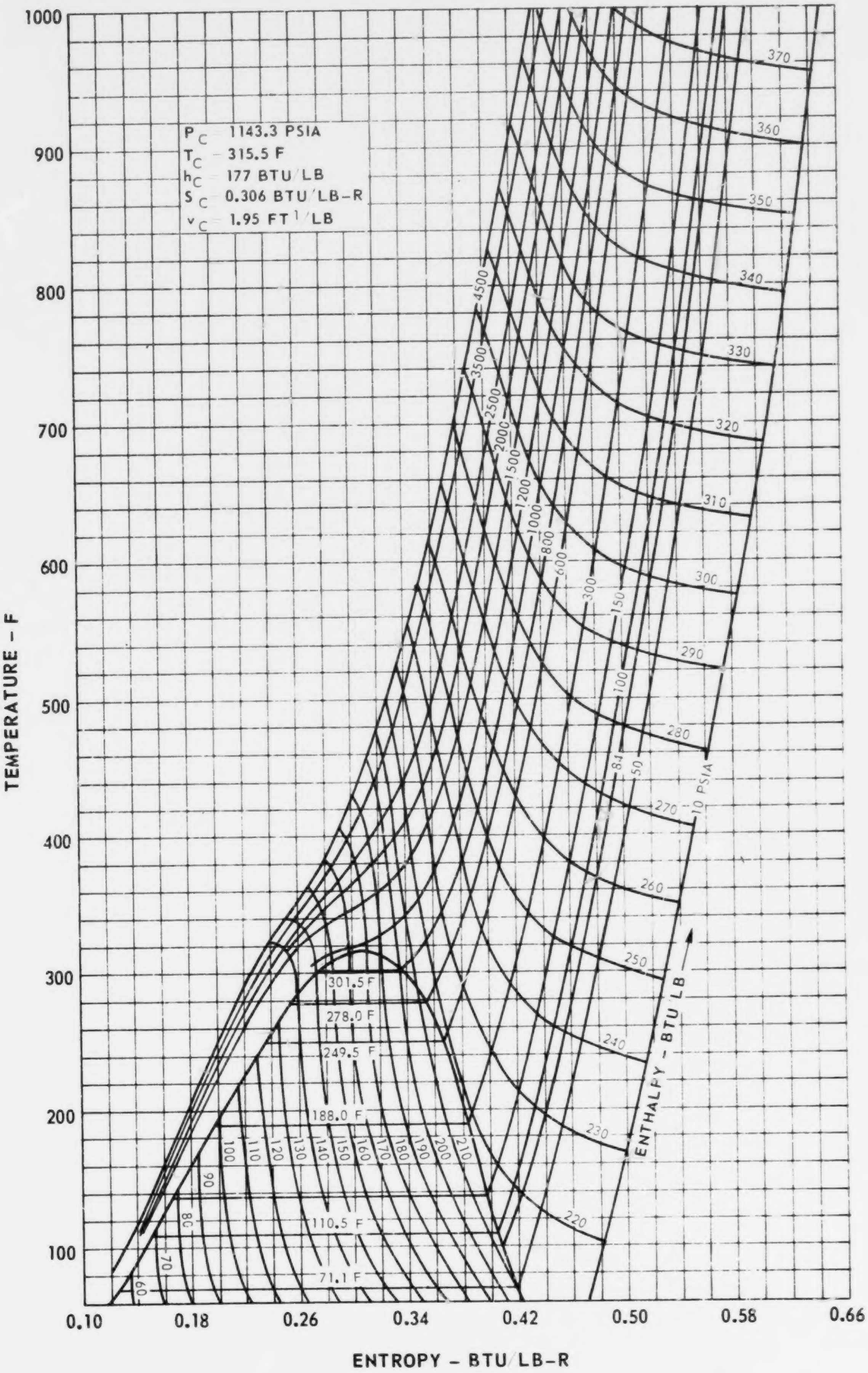
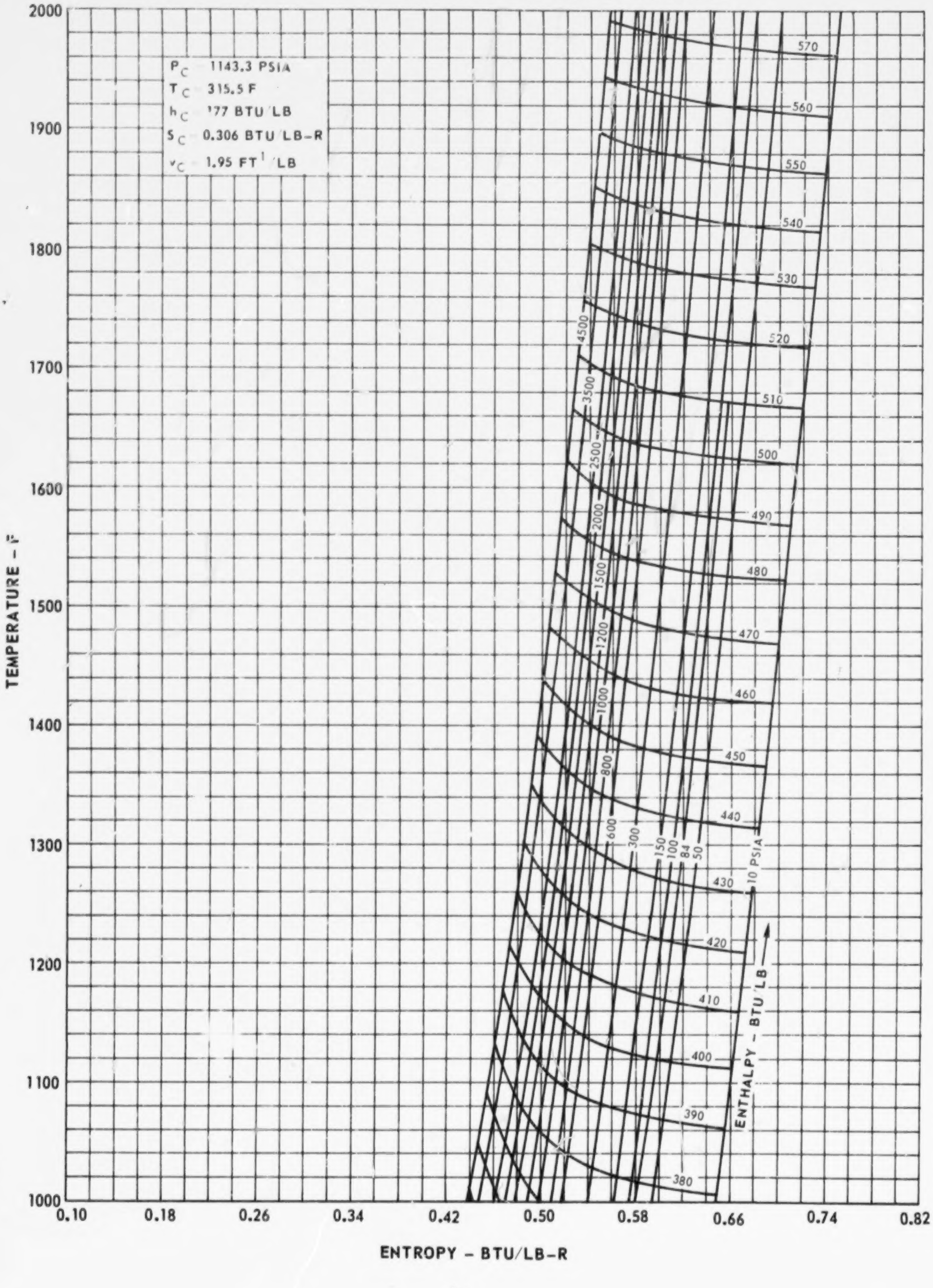
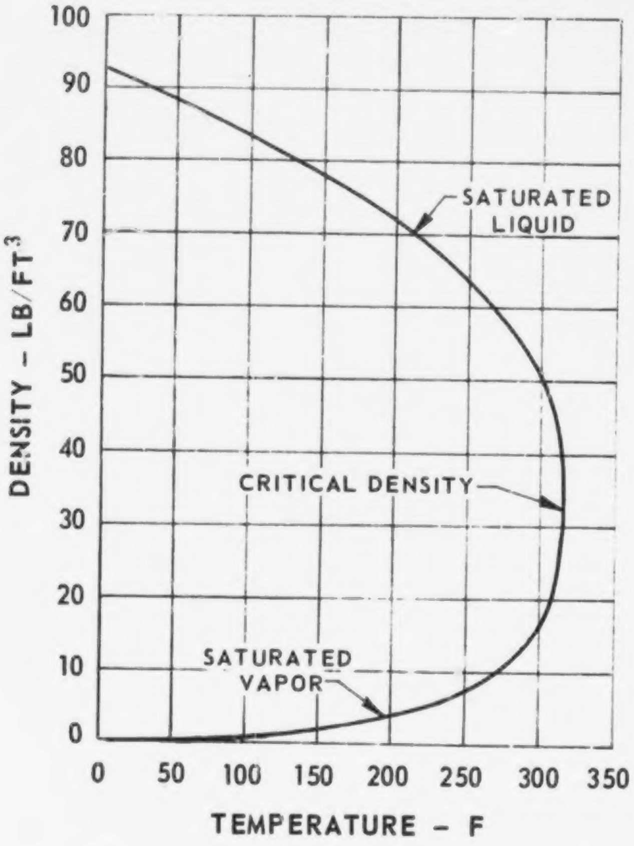


FIG. 193

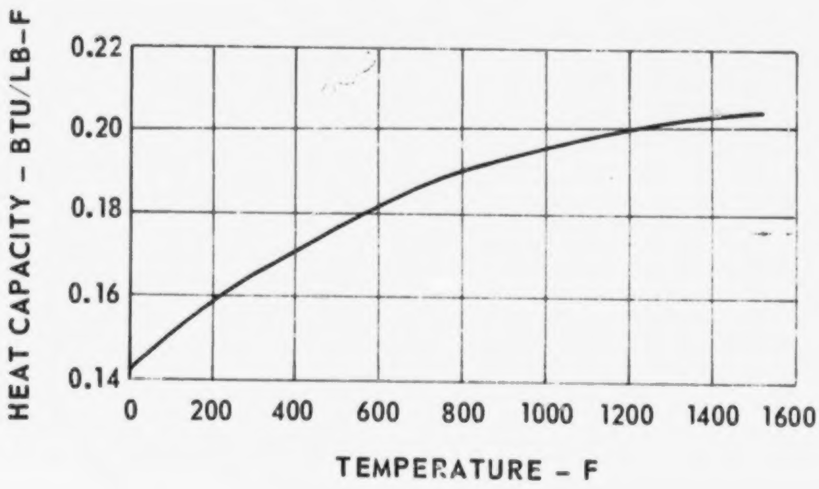
THERMODYNAMIC PROPERTIES OF SULFUR DIOXIDE



DENSITY AND HEAT CAPACITY OF SULFUR DIOXIDE



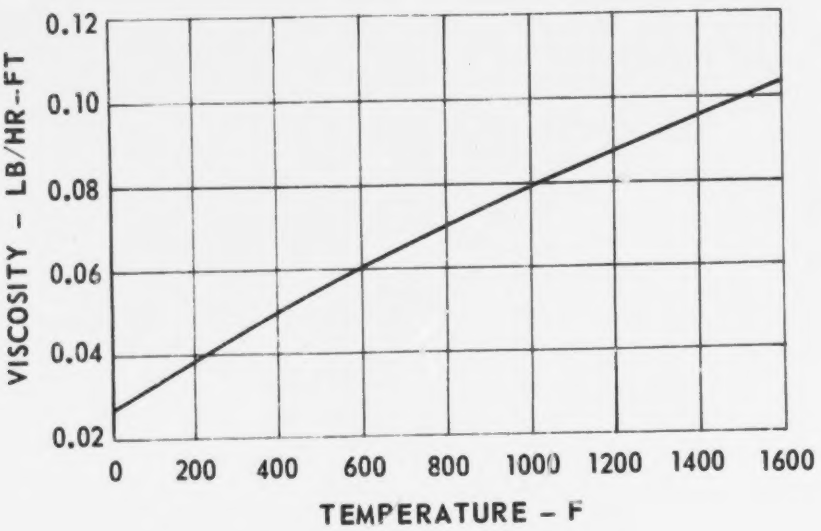
DENSITY OF SATURATED SULFUR DIOXIDE



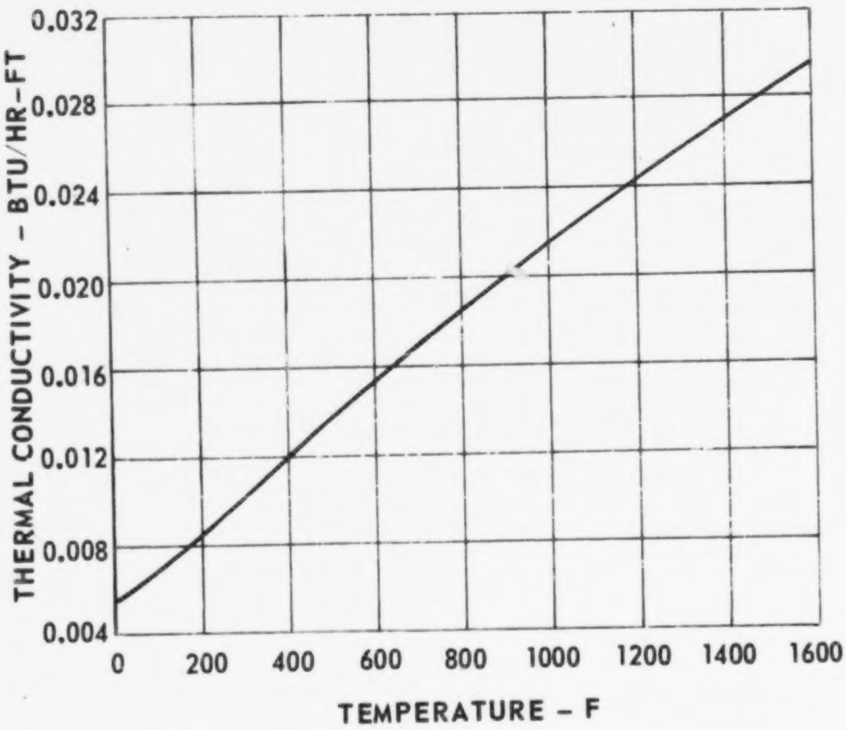
HEAT CAPACITY OF SULFUR DIOXIDE AT
ONE ATMOSPHERIC PRESSURE

FIG. 195

VISCOSITY AND THERMAL CONDUCTIVITY OF SULFUR DIOXIDE



VISCOSITY OF SULFUR DIOXIDE AT ONE ATMOSPHERIC PRESSURE



THERMAL CONDUCTIVITY OF SULFUR DIOXIDE AT ONE ATMOSPHERIC PRESSURE

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